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THERMODYNAMICS, ABRIDGED

THERMODYNAMICS, ABRIDGED

BASED ON "APPLIED THERMODYNAMICS FOR ENGINEERS" BY THE SAME AUTHOR

BY

WILLIAM D. ENNIS, M.E.

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PREFACE

(WHICH STUDENTS SHOULD READ)

This brief statement of fundamental principles has been prepared especially for the use of midshipmen in the United States Naval Academy. The work was undertaken at the request of Commander J. O. Richardson, U. S. N., Head of the Department of Marine Engineering and Naval Construction, and has been carried on in close coöperation with officers of that Department, particularly Comdr. W. L. Friedell, U. S. N., Lt. Comdr. T. W. Johnson, Professor of Mathematics, U. S. N., and Lt. Comdr. H. W. Boynton, U. S. N.

Thermodynamics is difficult, but worth while. To some extent, it has been simplified by planning the problems for easy solution. The table preceding Chapter II will be found useful for exponential expressions. The solution of many problems is necessary in order that a real grasp of the subject may be attained. All problems should be solved with the slide rule. This implies that answers will be absolutely reliable only with respect to two significant figures, the third figure being estimated. An error which may be as high as 1 per cent. is therefore allowable. The answers given have been obtained by slide rule, and are subject to this error. Other errors may occasionally be found during a first year's use of the book. The student's answer MAY be right, therefore, even when it disagrees with the answer in the book.

One per cent. accuracy is good enough for almost all practical engineering calculations. We rarely know the strength of a material with any greater exactness. Even the dimensions of structural parts are subject to a comparable error. Good engineering is largely a matter of NOT straining at gnats and swallowing camels.

If there is a "royal road" to any kind of learning, thermodynamics will be found to be the royal road to real comprehension of steam machinery.

NUMERICAL CONSTANTS AND COMMONER SYMBOLS

1.34 hp. = 1 kw.	746 watts = 1 hp.
2.3026 = $\log_e x \div \log_{10} x$.	778 ft. lb. = 1 B.t.u.
5.403 = 778 ÷ 144.	1544 = $PVm \div wT$, Art. 12.
32.17 = acceleration of gravity.	2545 B.t.u. per hr. = 1 hp.
42.42 B.t.u. per min. = 1 hp.	33 000 ft. lb. per min. = 1 hp.

1 980 000 ft. lb. per hr. = 1 hp.

B.t.u. = British thermal unit(s): Art. 3.
c = clearance: Arts. 7, 39.
C = compression ratio: Arts. 38, 149.
D = displacement: cu. ft: Arts. 7, 42.
e = efficiency: Arts. 34, 80.
= base of Napierian system of logarithms.
e_v = volumetric efficiency: Art. 42.
$E = T + I$ = internal energy of vapor: Art. 63.
F. = Fahrenheit.
F = factor of evaporation: Art. 73.
f = diagram factor: Art. 104.
g = 32.17 = acceleration of gravity.
h = heat of 1 lb. of boiling liquid: Art. 63.
H = total heat of 1 lb. of dry vapor: Art. 65.
H' = total heat of 1 lb. of superheated vapor: Art. 83.
H, H = heat received by a substance: Art. 2.
ΣH = net amount of heat received or rejected in a cycle: Art. 21.
hp. = horse power.
ihp., Ihp. = indicated horse power.
I = disgregation work: Art. 2.
I_0 = ice melting effect per ihp.-hr.: Art. 59.
kw. = kilowatt(s).
k = specific heat at constant pressure: Art. 14.
l = specific heat at constant volume: Art. 14.
L = stroke of an engine, ft.: Art. 7.
= latent heat of vaporization: Art. 64.
log = logarithm to the base 10.
$\log_e$ = logarithm to the base e = 2.3026 log.
m = molecular weight: Art. 12.
n = polytropic exponent: Art. 24.

n = entropy or change of entropy: Art. 68.
n_w = entropy of 1 lb. of liquid: Art. 71.
n_e = entropy of vaporization: Art. 71.
n_s = total entropy of dry vapor: Art. 71.
n' = total entropy of superheated vapor: Art. 83.
N = r.p.m.: Art. 7.
p = absolute pressure, lb. per sq. in. (= gauge pressure + 14.696).
P = absolute pressure, lb. per sq. ft. = 144 p.
p_m = mean effective pressure: Arts. 7, 39.
p_c = intercooler pressure: Art. 45.
Q = the quantity of heat in 1 lb. of a vapor: Art. 87.
R = $PV + wT$: Art. 12.
r = internal latent heat of vaporization: Art. 65.
= ratio of expansion: Arts. 106, 162.
r.p.m. = revolutions per minute.
s = specific heat: Arts. 3, 27.
t = temperature Fahrenheit.
T = $t + 460$ = absolute temperature: Art. 11.
T = temperature effect of heat: Art. 2.
t' = temperature of superheated vapor: Art. 83.
u, U = velocity, ft. per sec.: Arts. 56, 133.
v = volume of 1 lb. of a substance.
v' = volume of 1 lb. of a superheated vapor: Art. 83.
v_w = volume of 1 lb. of liquid: Art. 65.
V = volume of w lb. of a substance.
w = weight, lb.
W, W = external mechanical work done: Arts. 2, 26.
ΣW = net external work done in a cycle: Art. 21.
x = dryness fraction: Art. 74.
y = $k \div l$ = ratio of specific heats (Art. 17): also adiabatic exponent (Art. 28).

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THERMODYNAMICS, ABRIDGED

CHAPTER I

PRELIMINARY—GENERAL PRINCIPLES

1. **Subject-Matter.** Thermodynamics treats of heat as a source of motive power. All artificial motive powers (not excluding windmills, tide motors and muscular effort) originate from heat. Heat is not matter, but energy. It is measurable in foot-pounds. Heat may be regarded as due to molecular motion. Properly speaking, Thermodynamics is not concerned with phenomena of heat transmission, which belong in the field of general physics. Neither does it traverse the domain of thermochemistry, which deals with heats of formation without relation to mechanical effects. Thus our subject is from one standpoint merely a subdivision of the subject of heat, which latter constitutes one of the departments of physics. More broadly viewed, however, Thermodynamics deals with molecular mechanics while general physics (including practical astronomy) is concerned with mass mechanics.

Questions. How does muscular energy originate from heat? Show that the action of a windmill is due to heat. Show that the force which propels a steamship is derived from the heat of the sun.

2. **Effects of Heat.** Thermodynamics ignores electrical and chemical effects and (for the present) considers only three results following the application of heat to a substance:

(1) **Increase of Temperature.** This is the most familiar effect. If heat results from molecular motion, temperature may be regarded as due to molecular velocity. The molecules are

incessantly vibrating. The faster they move, the higher is the temperature. Temperature is measured by the thermometer and may be expressed in Fahrenheit or Centigrade units. Ice melts, normally, at 0° C. or 32° F.; water boils under standard conditions at atmospheric pressure at 100° C. or 212° F. A range of 100° C. means the same thing as a range of 180° F. Hence 1 degree range of Centigrade temperature = 9/5 degree range of Fahrenheit temperature, and

$$T_F = 9/5\ T_C + 32, \tag{1}$$

$$T_C = 5/9(T_F - 32). \tag{2}$$

Note that the Fahrenheit freezing point, 32°, enters into both equations.

(2) **Disgregation Work.** The molecules of fluids exert forces upon one another. When heat is supplied, the molecules may take up new positions, i.e., assume a new configuration. This requires that energy be expended to destroy the original equilibrium. Potential energy is stored up in establishing a new condition of equilibrium. The expenditure of energy necessary for rearranging the molecules is called disgregation work. Disgregation work occurs, for example, when ice is melted by heat.

(3) **External Mechanical Work.** In general, bodies expand when heated. The work done by reason of such expansion against resisting objects is called EXTERNAL work and is the effect of heat with which we are primarily concerned.

The action of heat may then be summed up thus: the heat, H, received by the substance is utilized, (*a*) in raising its temperature, (*b*) in rearranging its molecules, and (*c*) in performing mechanical work. In symbols,

$$H = T + I + W, \tag{3}$$

where T = heat expended in changing temperature,
I = heat expended in doing disgregation work,
W = heat expended in doing external work.

All terms must be expressed in the same unit, which may be

either a heat unit or a work unit. Signs may be either + or −. Thus if heat is EMITTED, the sign of H is −. The sign of T is − when the temperature FALLS. The sign of W is − if work is CONSUMED, i.e., if work is done ON the substance instead of BY it. When a substance EXPANDS, W is +. When it is COMPRESSED, W is −. Note that T is NOT the temperature, but the quantity of heat consumed in changing the temperature. It is not even the CHANGE of temperature. The sign of I is + if molecular forces have been overcome, as when a separation of mutually attractive molecules is effected. Its sign is − in the reverse instance, as when such molecules are brought closer together.

Prob. 1. At what temperature do the Centigrade and Fahrenheit thermometers read alike? (Ans., − 40°.)

Prob. 2. What is the (algebraic) amount of work done in rearranging the molecules if the heat received is 400, the heat represented by a rise of temperature 100 and the mechanical work done 120? (Ans., + 180.)

Prob. 3. If $W = -20$, $I = +30$, $T = +10$, find H. (Ans., + 20.)

Prob. 4. When 1 lb. of water at the boiling point is converted into steam, 970.4 units of heat must be supplied. The mechanical work done by the expansion of water into steam is 72.8 units. Compute T and I. (Ans., $T = 0$, $I = +897.6$.)

3. **Definitions and Laws.** Two bodies are at the same temperature when there is no tendency toward a transfer of heat from one to the other. When a heat-transfer occurs, heat always passes from a high-temperature body to one of lower temperature. The transfer tends to continue until the two temperatures are equal.

The HEAT UNIT (British thermal unit, B.t.u.) is the quantity of heat required to raise the temperature of 1 lb. of water 1° F. Since this quantity is somewhat variable according to the initial temperature, we will use the MEAN B.T.U., defined as 1/180 OF

THAT QUANTITY OF HEAT CONSUMED IN RAISING 1 LB. OF WATER FROM 32° F. TO 212° F.

The CALORIE is the quantity of heat required to raise 1 kilogram (2.2046 lb.) of water 1° C. (9/5° F.). Its value is

$$2.2046 \times \frac{9}{5} = 3.9683 \text{ B.t.u.}$$

The SPECIFIC HEAT of a substance is the quantity of heat (number of heat units) necessary to increase the temperature of 1 lb. of that substance 1° F. By definition, then, the mean specific heat of water from 32° F. to 212° F. is 1.0. The specific heat of a gas is not an absolute constant (see Art. 14).

When H B.t.u. are supplied to w lb. of a substance having a specific heat s,

$$H = ws(T_2 - T_1), \tag{4}$$

where T_2 and T_1 are the final and initial temperatures respectively.

When several fluids are mixed, some will lose heat and others will gain heat. The total loss (including loss of heat by any surrounding bodies) must equal the total gain. In other words, the algebraic sum of heat losses is zero:

Loss by first body + loss by second body

+ loss by third body $\cdots$ + radiation "loss" = 0

$$w_a s_a(T_a - T_m) + w_b s_b(T_b - T_m) + w_c s_c(T_c - T_m) \cdots + H_R = 0. \tag{5}$$

Here w_a, w_b, w_c, denote weights of the several fluids in lb.,

s_a, s_b, s_c, their respective specific heats,

T_a, T_b, T_c, their respective initial temperatures,

T_m, their common final temperature,

H_R, the heat contributed by surrounding bodies (which will be negative if T_m exceeds the surrounding temperature).

Prob. 5. How many (mean) B.t.u. are required to heat 4 lb. of water from 32° to 212°? (Ans., 720.)

Prob. 6. If 1 meter = 39.37 in., how many kilogram-meters are equivalent to 1 ft. lb.? (Ans. 0.1383.)

Prob. 7. If the specific heat is 0.10, how many calories are consumed in heating 10 lb. 100° F.? (Ans., 25.2.)

Prob. 8. When 1 lb. of water is heated from 100° to 200°, 99.97 B.t.u. are expended. What is the mean specific heat of water for this range? (Ans., 0.9997.)

Prob. 9. The specific heat is 0.3, the weight $3\frac{1}{3}$ lb. How many B.t.u. are consumed if the temperature rises 500° F.?

(Ans., 500 B.t.u.)

Prob. 10. A mixture is made of three liquids, the weights being 5, 3 and 22 lb., the corresponding specific heats 1.0, 0.3 and 0.12 and the final temperature 189°. The original temperatures were 200°, 110° and 220°. How much heat was lost by radiation during the mixing? (Ans., $H_R = -65.7$ B.t.u.)

4. **Mechanical Equivalent of Heat.** The "first law" of Thermodynamics is,

> HEAT AND MECHANICAL ENERGY ARE MUTUALLY CONVERTIBLE, IN THE RATIO OF 778 FT. LB. (777.5 MORE NEARLY) TO 1 B.T.U.

This law has been established by direct experiment (expending work in agitating a fluid and measuring the heating accomplished) and from abundant inference. 1 B.t.u. = 778 ft. lb., just as 1 yard = 3 ft. We do not always realize 778 ft. lb. of useful work from 1 B.t.u., because heat does not equal W, but $T + I + W$ (Equation 3). The number 778 is called the MECHANICAL EQUIVALENT OF HEAT.

In a heat engine test, each pound weight of steam entered the cylinder containing 1125 B.t.u. Leaving the cylinder, each pound contained 1000 B.t.u. Then 125 B.t.u. disappeared. The engine developed 150 horsepower = 150 × 33 000 or 4 950 000 ft. lb. per min. It used 55 lb. of steam per min., or 55 × 125 = 6875 B.t.u., if we assume the heat which disappeared to have been "used." Then 6875 B.t.u. = 4 950 000

ft. lb. or 1 B.t.u. = 720 ft. lb. This is as close an approximation to the true value as might be expected from this type of experiment.

Prob. 11. How many kilogram-meters are equivalent to 1 calorie (see Prob. 6). (Ans., 427.)

Prob. 12. 1 lb. of coal contains 14 000 B.t.u. If this could be fully utilized, how much coal would be burned per min. to develop 1 hp.? (Ans., 0.00303 lb.)

Prob. 13a. If a 100 hp. engine operates so as to cause the disappearance of 4250 B.t.u. per min., what is the probable value of the mechanical equivalent of heat? (Ans., 778.)

5. **Efficiency.** Efficiency is in general EFFECT ÷ CAUSE or OUTPUT ÷ INPUT. In Thermodynamics, efficiency is MECHANICAL WORK DONE ÷ TOTAL HEAT RECEIVED. Units must be alike. A HORSE POWER (hp.) is 33 000 ft. lb. per min. or 1 980 000 ft. lb. per hr. It is also 42.42 B.t.u. per min. or 2545 B.t.u. per hr. At 100 per cent. plant efficiency, an engine of 1 hp. would consume 2545 B.t.u. (IN THE COAL) per hr. At 10 per cent. efficiency it would consume 25 450 B.t.u. The efficiency of a mechanical device (a chain hoist, for example) may be increased indefinitely by fine fitting, ball bearings, etc., with 100 per cent. as the ultimate limit. Efficiencies of heat engines have a much lower (though for given conditions, a perfectly definite) limit. The actual efficiency of a given engine will always be below this limit and will vary with the conditions according to laws which Thermodynamics aims to discover. Thermodynamics shows how big to make an engine and how efficient it is likely to be, and why.

Prob. 13b. A ship uses 2 lb. of coal per hr. per indicated hp. The heat value of the coal is 12 725 B.t.u. per lb. What is the efficiency from fuel to cylinder? (Ans., 0.10, or 10 per cent.)

Prob. 14. An oil engine has an efficiency of 0.13, or 13 per cent. It develops 100 hp. using oil of 19 000 B.t.u. per lb. What weight of oil will it consume per hr.? (Ans., 102 lb.)

6. **Friction: Traction: Plant Efficiency.** Thermodynamics is concerned essentially with the conversion of HEAT IN STEAM to WORK IN THE CYLINDER of the engine. The corresponding efficiency, E_T, is called the THERMAL EFFICIENCY. Mechanical friction makes the work at the shaft or "brake" less than that done in the cylinder. The ratio of shaft hp. to cylinder hp. is E_M, the MECHANICAL EFFICIENCY. The efficiency from steam to shaft is $E_T \times E_M$. The efficiency of a series of conversions is always the product of the efficiencies of the individual conversions: $E = E_1 \times E_2 \times E_3 \cdots$. In a turbine, there is no "cylinder" and the only efficiency determinable is the ratio of work at the shaft to heat supplied by the steam.

If the resistance of a ship or railway train or other propelled object is R lb., the actual hp. expended in propulsion is

$$\frac{RV}{550} = \frac{RV_M}{375} = \frac{RV_K}{326},$$

where V, V_M and V_K are speeds in ft. per sec., miles per hr. and knots per hr., respectively.

Prob. 15. The shaft hp. is 80, the horse power lost in friction is 20 and 2 545 000 B.t.u. are consumed per hr. Find E_T and E_M. (Ans., 0.10, 0.80.)

Prob. 16. In an electric central station, 2 lb. of coal (14 000 B.t.u. per lb.) are used per kw. hr. (1 kw. = 1.34 hp.). Efficiencies are as follows: fuel to steam, 0.75: steam to work in cylinder, 0.25: shaft to switchboard output, 0.8. Find efficiency from cylinder to shaft. (Ans., 0.81.)

Prob. 17. The battleship PENNSYLVANIA has a resistance of 254 000 lb. at 20 knots. If $E_M = 0.80$ and the efficiency of the propellors is 0.70, what horse power is developed in the cylinders? (Ans., 27 700.)

7. **Engine Action.** Fig. 1 shows the piston P moving in the cylinder C and communicating via the piston rod R with the external mechanism. Inlet and outlet passages, I and E, are provided with valves, not shown. The piston being in its ex-

treme left-hand position (i.e., as far to the left as the external mechanism will permit), the distance a is called the LINEAR CLEARANCE. The volume then included between the left face

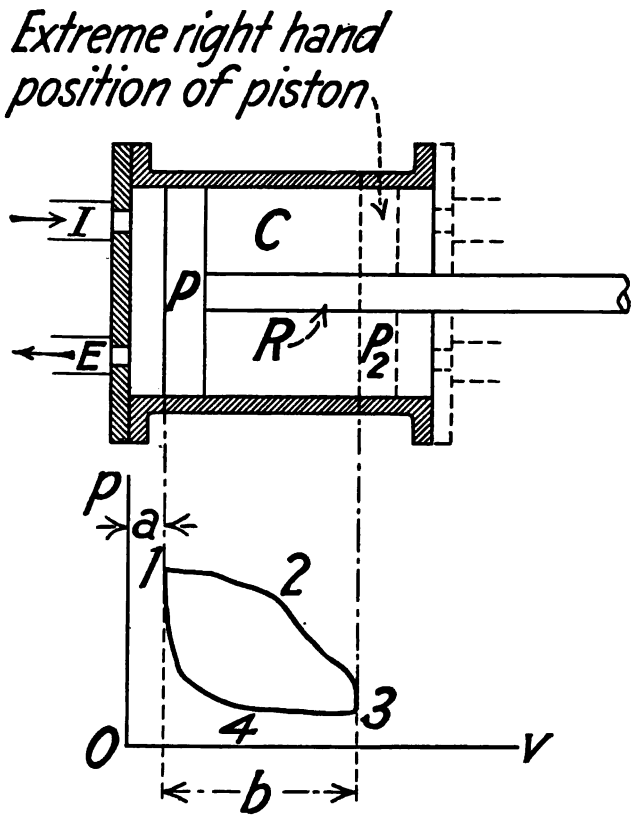

FIG. 1. Action of Engine.

of the piston and the closed faces of the valves is the CLEARANCE VOLUME, v_c. The piston now makes one full stroke to the right. The length of stroke, or distance moved, is determined by the external mechanism. If the whole volume swept through by the piston in such a stroke is D, then the CLEARANCE (proportion of clearance) is $c = v_c/D$. If d = piston diameter, L = piston stroke, $D = (\pi/4)d^2L$. The quantity D is called the DISPLACEMENT PER STROKE.

Most steam engines have the right-hand end of the cylinder closed by a head with inlet and outlet passages. The rod then passes through a stuffing box in the head, not shown. The engine is then called DOUBLE-ACTING, steam being used on both sides of the piston. Most gas and oil engines are SINGLE-ACTING. In single-acting engines, the right-hand end of the cylinder is left open. This gives a better opportunity for cooling the piston and piston rod without circulating cold water through them. In double-acting engines there is of course a clearance at each end of the cylinder.

The INDICATOR depicts on paper the pressure existing at various points (piston positions) during the stroke. The lower part of Fig. 1 shows a record of this kind, called an INDICATOR DIAGRAM. Ordinates represent pressures on the left-hand side of the piston, abscissas represent piston movements. The upper line 123 represents the pressures applied while the piston is moving to the right. The lower line 341 represents RESISTING pressures during the return (left-hand) stroke. The diagram shows what occurs on the left-hand side, only, of the piston. It records two strokes. Under constant conditions, subsequent records would simply duplicate this one.

Abscissas (which represent piston movements) also represent volumes (to another scale), since the cylinder diameter is the same at all points. The volume swept through = area of piston $\times$ distance piston has moved. The vertical axis (of zero volume) falls directly below the inner face of the left-hand cylinder head ONLY when the valve faces are flush with that face. This is a condition which rarely if ever occurs. The *ov* axis is located at zero absolute pressure, i.e., 14.696 lb. per sq. in. below the pressure of the atmosphere. Obviously $c = a \div b$. In a double-acting engine, a symmetrically disposed oppositely placed diagram represents the action on the right-hand side of the piston.

The average ordinate of the indicator diagram (= area $\div$ b, with proper consideration of scale) is p_m, the MEAN EFFECTIVE PRESSURE, usually taken in lb. per sq. in. It is the average

pressure exerted on one side of the piston during one stroke, the pressure during the succeeding stroke being regarded as zero. In a double-acting engine, however, there is an approximately equal average pressure on the other side of the piston during this succeeding stroke: hence the average pressure p_m is maintained continuously. If d = piston diameter in inches, the average total pressure on the piston (disregarding the reduction of area on one side by the rod) is $(\pi/4)d^2p_m$ lb. If L = stroke in ft., A = piston area in sq. in. = $(\pi/4)d^2$, and the engine makes N r.p.m., the pressure p_m acts through a distance of L ft. per stroke, $2L$ ft. per revolution and $2LN$ ft. per min., and the cylinder horse power is

$$\text{ihp.} = (\pi/4)d^2p_m \times \frac{2LN}{33\,000} = \frac{\pi p_m L d^2 N}{66\,000}$$

$$= \frac{p_m AS}{33\,000} = \frac{2p_m ND}{12 \times 33\,000} = \frac{2p_m LAN}{33\,000}, \quad (6)$$

where S = PISTON SPEED = $2LN$ and $D = (\pi/4)d^2L \times 12$.

The OVERLOAD CAPACITY of an engine is (MAXIMUM POWER — NORMAL POWER) ÷ NORMAL POWER. If p_m is the same at all powers,

$$\text{overload capacity} = \frac{N' - N}{N}, \quad (7)$$

where N = normal speed and N' = maximum speed. If N is constant for all powers,

$$\text{overload capacity} = \frac{p_m' - p_m}{p_m}, \quad (8)$$

where p_m is the normal value and p_m' the maximum value. In most engines, p_m varies. In marine engines, both p_m and N vary.

The mean effective pressure fully determines the power of an engine of given size and speed.

Prob. 18. In an engine 10 in. diameter by 20 in. stroke the linear clearance is $\frac{1}{4}$ in. Find the displacement per stroke in cu. in. If the valve faces are flush with the inner face of the

cylinder head, what is the clearance volume? The proportion of clearance? (Ans., 1570.8: 19.635 cu. in.: 0.0125 or $1\frac{1}{4}$ per cent.)

Prob. 19. The clearance volume of a 10 by 20 (diameter by stroke) in. engine is found to hold 58.905 cu. in. of water. What is its clearance? (Ans., $3\frac{3}{4}$ per cent.)

Prob. 20. The indicator diagram from a 10 by 20 in. engine is 3.1416 in. long. What displacement is represented by 1 in. of diagram length? (Ans., 500 cu. in.) If the clearance of this engine is 5 per cent., what will be the dimension a, in Fig. 1? (Ans., 0.15708 in.)

Prob. 21. The area of an indicator diagram is 3 sq. in. and its length is 4 in. If the vertical scale is such that 1 in. represents 40 lb. per sq. in., what is the mean effective pressure? (Ans., 30 lb. per sq. in.)

Prob. 22. In a 10 by 20 in. double-acting engine at 100 r.p.m., with $p_m = 30$, find the piston speed and ihp. (Ans., $333\frac{1}{3}$ ft. per min., 23.8 hp.)

Prob. 23. A constant-speed 10 by 20 in. engine has a normal $p_m = 30$ and a maximum $p_m' = 60$. What is its overload capacity? (Ans., 1.0 or 100 per cent.)

Prob. 24. At 100 r.p.m., $p_m = 30$. At 300 r.p.m., $p_m = 24$. Find overload capacity if 100 r.p.m. is the normal speed. (Ans., 140 per cent.)

Prob. 25. An engine rated at 100 hp. has an overload capacity of 210 per cent. What is the maximum hp. it can develop? (Ans., 310 hp.)

VALUES OF EXPONENTIALS IN CHAPTERS II AND III

n	$\frac{1}{n}$	$\frac{n-1}{n}$
1.3	0.769	0.231
1.35	0.74	0.259
1.4	0.714	0.286
1.406	0.713	0.289

$0.355^{0.231}$	$= 0.787$	$3.5^{0.74}$	$= 2.52$
$0.9^{0.74}$	$= 0.925$	$4^{0.259}$	$= 1.43$
$(\frac{10}{8})^{3.5}$	$= 6$	$4^{0.289}$	$= 1.49$
$2^{7/6}$	$= 2.24$	$4^{0.74}$	$= 2.78$
$3^{1.406}$	$= 4.68$	$10^{-0.289}$	$= 0.514$
$3.04^{0.74}$	$= 2.28$	$10^{0.286}$	$= 1.93$
$3.37^{0.259}$	$= 1.37$	$10^{0.289}$	$= 1.95$
$3.37^{0.74}$	$= 2.46$	$10^{0.714}$	$= 5.18$
$3.39^{0.259}$	$= 1.38$	$10^{0.74}$	$= 5.50$
$3.39^{0.74}$	$= 2.47$	$11.08^{0.259}$	$= 1.86$
$3.5^{0.259}$	$= 1.38$	$11.08^{0.74}$	$= 5.93$

CHAPTER II

PERMANENT GASES. LAWS AND PROCESSES

General Statement

A clear understanding of the laws governing the behavior of permanent gases is absolutely essential to a proper comprehension of steam machinery. Although those laws may seem for the next few pages to have little bearing on the heat-power equipment of a ship, their application will soon be shown. The few paragraphs which follow are as important in steam engineering as the multiplication table is in arithmetic.

8. **Gases.** Thermodynamics deals, generally, with fluids rather than with solids, and (more narrowly) considers vapors and gases rather than liquids. A vapor is formed when a liquid evaporates. If the temperature of a vapor is greatly increased it becomes what we call a gas. Steam at 1200° is a gas. Air exists in the liquid form at about − 300° F. It might be considered a vapor at − 250° F. In ordinary human experience, it is a gas. The present discussion deals with the "permanent" gases. Dry air, hydrogen, nitrogen, oxygen and a few of the compound gases are of this variety. Steam, NH_3, CO_2 and SO_2, on the other hand, are to be regarded as vapors. They will be examined later. In many thermodynamic calculations, great simplicity results (without any serious sacrifice of accuracy) from considering the permanent gases to be "perfect," i.e., to conform EXACTLY to certain physical laws which in reality are only approximated.

The properties of gases which are chiefly to be considered are the PRESSURE, VOLUME and TEMPERATURE. By pressure (p, lb. per sq. in.) we are to understand the uniform absolute (NOT gauge) pressure exerted per unit of surface by the gas on surrounding objects or by surrounding objects on the gas. The

volume, v, is taken in cu. ft. The symbol t represents the uniform Fahrenheit temperature of the gas.

9. **Boyle's Law.** IF THE TEMPERATURE OF A GAS IS KEPT CONSTANT, THE PRESSURE VARIES INVERSELY AS THE VOLUME.

$$\frac{p_1}{p_2} = \frac{v_2}{v_1} \quad \text{or} \quad p_1v_1 = p_2v_2 \quad \text{or} \quad pv = \text{const. when } t \text{ is const.} \quad (9)$$

This is the equation of an equilateral hyperbola, referred to its asymptotes. When $p = 14.696$ and $t = 32°$, v for 1 lb. of air is 12.387: hence at this temperature and for 1 lb. of air pv is ALWAYS equal to $14.696 \times 12.387 = 182.04$. Fig. 2 represents Boyle's law, the various curves being plotted for various values of the constant.

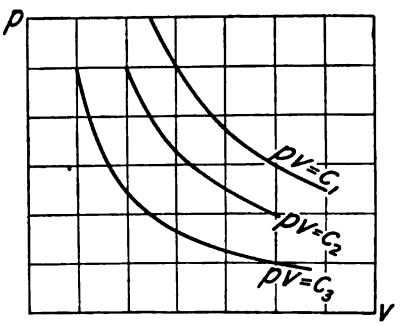

FIG. 2. Boyle's Law.

Note that at sea level and normal barometer, $p =$ GAUGE PRESSURE + 14.696. For ordinary calculations, 14.7 is near enough.

Prob. 26. What is the volume of 5 lb. of air at 32° F. if the pressure is 10 atmospheres? (Ans., 6.1935 cu. ft.)

Prob. 27. If the volume of 1 lb. of hydrogen is 100 cu. ft. when its pressure is 5.304 lb. by gauge, what will be the volume of 4 lb. of the same gas at the same temperature when its gauge pressure is 15.304 lb.? (Ans., $266\frac{2}{3}$ cu. ft.)

Prob. 28. The gauge pressure on a gas tank is 85.304 lb., and its contents weigh 50 lb. What weight will it hold of the same gas at the same temperature under a gauge pressure of 185.304 lb.? (Ans., 100 lb.)

10. **Charles' Laws.** (*a*) IF THE VOLUME OF A GAS IS KEPT CONSTANT, EQUAL INCREMENTS OF PRESSURE ACCOMPANY EQUAL

INCREMENTS OF TEMPERATURE. Fig. 3 illustrates this. At constant volume, the relation between pressure and temperature is represented by a straight line. The equation is, $p = at + b$, where a and b are constants.

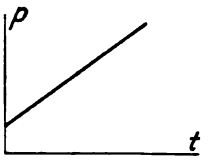

FIG. 3. Charles' Law. (Constant Volume.)

(*b*) IF THE PRESSURE OF A GAS IS KEPT CONSTANT, EQUAL INCREMENTS OF VOLUME ACCOMPANY EQUAL INCREMENTS OF TEMPERATURE. Again there is a straight line law: $v = a_1t + b_1$: Fig. 4.

Prob. 29. The pressure in a closed tank is 200 lb. (absolute) at 32°, 400 lb. at 524°. What will it be at 770°? (Ans., 500 lb.)

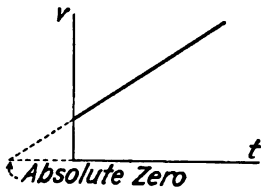

FIG. 4. Charles' Law. (Constant Pressure.)

Prob. 30. The absolute pressure of the gas in a balloon is 15 lb. at 40° and 30 lb. at 540°. If we disregard any increase in volume of the balloon, what is the pressure at 90°? (Ans., 16.5 lb.)

11. **Absolute Zero.** In Fig. 4, suppose the gas to be cooled indefinitely while conforming with the straight line law. At some very low temperature, the volume will then become zero. Experiments indicate a contraction of 1/492 of the original volume per degree of cooling if the gas is originally at 32° F. Hence zero volume would be reached after 492° of cooling, or at a temperature of − 460° F. This temperature (more accurately − 459.6° F.) is called the ABSOLUTE ZERO. Experiments on the reduction of pressure at constant volume indicate the same point for the absolute zero. Hence Charles' laws may be written,

$$\left.\begin{aligned} \frac{p_1}{p_2} &= \frac{T_1}{T_2} \quad \text{when } v \text{ is constant} \\ \frac{v_1}{v_2} &= \frac{T_1}{T_2} \quad \text{when } p \text{ is constant} \end{aligned}\right\} \tag{10}$$

Here $T = t + 460$ and denotes (as always hereafter) the ABSOLUTE TEMPERATURE.

No actual gas conforms exactly with the laws of Boyle and Charles. Zero volume would never be actually realized. The laws are close approximations for the behavior of "permanent" gases under normal conditions.

Prob. 31. What is the location of the absolute zero on the Centigrade scale? (Ans., — 273° C.)

Prob. 32. If an automobile tire did not expand, what would be the percentage increase in absolute pressure when the temperature rose from 40° to 90° F.? (Ans., 10 per cent.)

Prob. 33. The pressure in a closed tank is 200 lb. absolute when $t = 32°$. What is the pressure at 524°? Note Prob. 29. (Ans., 400 lb.)

12. **The Perfect Gas: Its Law.** Conceive of a gas which conformed rigorously with the laws of Boyle and Charles (Arts. 9, 10, 11). Such a hypothetical substance is called the PERFECT GAS. In many calculations, for dry air and its constituents and for hydrogen, these gases are treated as perfect, with a close approach to correctness. The behavior of the perfect gas may be expressed by a single equation which combines the laws already given.

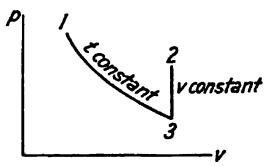

FIG. 5. Law of a Perfect Gas.

In Fig. 5, let the points 1 and 2 represent any two states of a given weight of gas. Draw an equilateral hyperbola 1–3 and a vertical line 3–2, intersecting at some point 3. Along 1–3, t is constant and $p_1v_1 = p_3v_3$. Along 2–3, v is constant and $p_2/p_3 = T_2/T_3$, or $p_3 = p_2(T_3/T_2)$. Then (since $v_3 = v_2$), $p_1v_1 = p_3v_3 = p_2v_2(T_3/T_2)$ or (since $T_3 = T_1$),

$$\frac{p_1v_1}{T_1} = \frac{p_2v_2}{T_2} \quad \text{or} \quad \frac{pv}{T} = \text{constant.}$$

If pv/T = const., then pv = const. when T is const. — Boyle's law. If $p_1v_1/T_1 = p_2v_2/T_2$, then when p is const. $v_1/v_2 = T_1/T_2$ — Charles' law.

For air, when $p = 14.696$ and $t = 32°$, $T = 492°$ and v per lb.

is 12.387. Usual units employed are $P = 144p$, the pressure in lb. per sq. ft. Then $Pv/T = (144 \times 14.696 \times 12.387) \div 492 = 53.36$ or $Pv = 53.36T$. For w lb. having a volume of V cu. ft., $PV = 53.36wT$. For any gas,

$$PV = wRT, \tag{11}$$

in which R is some constant. This is the EQUATION OF THE PERFECT GAS.

$P = 144p$ = absolute pressure, lb. per sq. ft.,

V = volume, cu. ft.,

w = weight, lb.,

T = absolute temperature = $t + 460$.

Equation (11) includes three variables. It is therefore the equation of a SURFACE. A perspective representation of this surface appears in Fig. 5*a*. Any plane perpendicular to the OT axis cuts an equilateral hyperbola (like *psxv*) in the surface, the plane being a locus of constant temperature and the hyperbola representing Boyle's law. Charles' law for constant pressure is indicated by the straight lines *map*, *ls*, cut in the surface by planes perpendicular to the OP axis. If planes are drawn perpendicular to the OV axis, the intersections with the surface are straight lines *xy*, *vben*, which represent the other form of Charles' law.

Values of R for permanent gases are found as follows. Put a gas under "standard" conditions of pressure (14.696 lb. per sq. in. absolute) and temperature (32° F., or 491.6° absolute) and measure the volume of 1 lb. Call this volume v_0. Then

$$R = \frac{PV}{wT} = \frac{144 \times 14.696v_0}{1 \times 491.6} = 4.305v_0.$$

At any fixed condition, however, THE VOLUMES OF GASES ARE INVERSELY PROPORTIONAL TO THEIR MOLECULAR WEIGHTS, or $v_0 = c \div m$, where c is a number which has the same value for all gases, and m = molecular weight. The value of c may be determined from the volume and molecular weight of any gas.

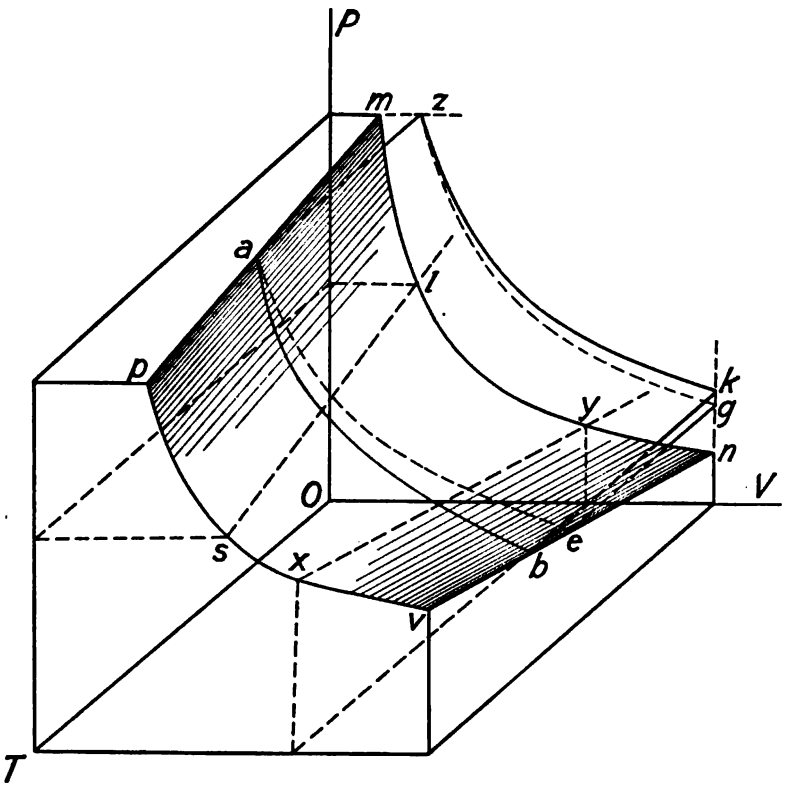

FIG. 5a. Thermodynamic Surface for a Perfect Gas.

Thus, for hydrogen (molecular weight = 2.016), v_0 is known to be $1 \div 0.00562$. Then c (based on standard conditions, as above defined) is $2.016 \div 0.00562 = 358.7$. For any permanent gas, $v_0 = 358.7 \div m$. The value of R for any gas is then $(4.305 \times 358.7) \div m = 1544 \div m$. For hydrogen, $R = 1544 \div 2.016 = 766$. The expanded form of the perfect gas equation is then

$$PV = \frac{1544wT}{m}. \tag{12}$$

The perfect gas is one having inert molecules: i.e., molecules which neither attract nor repel one another. Permanent gases approach this condition.

Prob. 34. An automobile tire carries 100 lb. gage pressure at 40° F. and expands 5 per cent. when heated from 40° to 90° F. What is the pressure after heating? (Ans., 105.3 lb., gage.)

Prob. 35. A balloon filled with hydrogen at 15 lb. per sq. in. absolute pressure has a volume of 766 000 cu. ft. at 80° F. What weight of gas does it hold? What weight of air at 14.696 lb. pressure and 70° F. does it displace? When the volume increases 10 per cent. and the temperature of the gas is 90° what is the pressure?

(Ans., 4000 lb.; 57 400 lb.; 13.9 lb. per sq. in. absolute.)

Prob. 36. What is the temperature of air at 1000 atmospheres pressure (i.e., $P = 1000 \times 14.696 \times 144$) when it weighs 80.75 lb. per cu. ft.? (Ans., 32° F.)

Prob. 37. What is the ratio of sizes of tanks for an oxy-hydrogen light if both tanks have like pressures and temperatures? Note $2H_2 + O_2 = 2H_2O$. Take molecular weight of oxygen at 32. (Ans., the oxygen tank is 50.4 per cent. the size of the hydrogen tank.)

Prob. 38. A tank holds 560.4 lb. of nitrogen (molecular weight 28.02) at a pressure of 100 000 lb. per sq. ft. and a temperature of 40° F. What is its volume? (Ans., 154.4 cu. ft.)

Prob. 39. A 44 cu. ft. tank holds 10 lb. of gas at 794° F. and at a pressure of 10 000 lb. per sq. ft. What is the molecular weight of the gas? (Ans., 44.)

Prob. 40. In the previous problem, after using gas from the tank the pressure is found to be 4400 lb. per sq. ft. and the temperature 140° F. What weight of gas has been removed?

(Ans., 0.8 lb.)

13. **Mixtures of Gases: Dalton's Law.** Excluding chemical actions, THE PRESSURE OF A MIXTURE OF GASEOUS BODIES IS THE SUM OF THE PRESSURES AT WHICH THE VARIOUS CONSTITUENTS WOULD EXIST IF EACH IN TURN ALONE OCCUPIED THE SAME SPACE AS THE MIXTURE AT THE SAME TEMPERATURE. Let the

mixture condition be P, V, T, and let subscripts denote constituents. Then Dalton's law is

$$P = P_1 + P_2 \cdots = \frac{R_1 T w_1}{V} + \frac{R_2 T w_2}{V} \cdots$$

$$= \frac{T}{V}(R_1 w_1 + R_2 w_2 \cdots) = \frac{RTw}{V} \quad (13)$$

where $w = w_1 + w_2 \cdots$ and R refers to the whole mixture. The equivalent value of R in the equation $PV = wRT$ is thus

$$(R_1 w_1 + R_2 w_2 \cdots) \div (w_1 + w_2 \cdots).$$

Prob. 41. What are the values of R for oxygen and nitrogen? (Ans., 48.25, 55.1.)

Prob. 42. Considering air to be 3 parts oxygen to 10 of nitrogen, by weight, what should be the value of R for air? (Ans., 53.5.)

Prob. 43. In the preceding, what part of standard atmospheric pressure (14.696 lb. per sq. in.) is due to the oxygen? (Ans., the pressure due to oxygen is 26.3 per cent. of that due to nitrogen and the oxygen pressure is 3.06 lb. per sq. in.)

14. **Specific Heats of Gases.** A gas may absorb heat while remaining at constant temperature (Art. 20), the specific heat being then infinite; for specific heat is heat divided by CHANGE of temperature: or it may change in temperature without absorbing heat (Art. 28), when the specific heat is zero. That this must sometimes be the case is evident from the fundamental equation, $\boldsymbol{H} = \boldsymbol{T} + \boldsymbol{I} + \boldsymbol{W}$. If $\boldsymbol{H}$ is + and ($\boldsymbol{W} + \boldsymbol{I}$) is + and greater than $\boldsymbol{H}$, $\boldsymbol{T}$ must be NEGATIVE. We cannot assign numerical values for the specific heats of gases unless at the same time we stipulate the type of process with which such specific heats are realized.

Two standard processes are clearly recognized. The gas may be kept at CONSTANT PRESSURE. The corresponding specific heat, called k (value 0.2375 for air), will then be fairly constant over ordinary temperatures. Second, the gas may be kept at

CONSTANT VOLUME. In this case, the (reasonably) constant specific heat is called l, the value for air being 0.1689. At very high temperatures, k and l are rapidly variable.

15. **Difference of Specific Heats.** When a gas is kept at CONSTANT VOLUME, it can do no mechanical work, for there is no such work possible without mass-motion. If heated at constant volume, energy will be stored up: so will it be stored when the gas is raised in temperature at constant pressure. If at constant pressure mechanical work is done, then the difference between k and l is due wholly to the performance of work in the one case and not in the other, and k will be greater than l.

16. **Work Done at Constant Pressure.** Mechanical work is done when a FORCE acts over a DISTANCE. If the force is in lb. and the distance in ft., the work, in ft. lb., is FORCE $\times$ DISTANCE. Take the case of a gas, at the pressure P_1, having a free (movable) surface of A sq. ft. Let expansion occur in a direction perpendicular to such surface, and let the linear movement in this direction be L ft. The volume of the gas must increase during the movement, by an amount equal to the space swept through by the moving surface. This space is AL cu. ft. If V_2 and V_1 are final and initial volumes in cu. ft., then $AL = V_2 - V_1$ and $W = P_1AL = P_1(V_2 - V_1)$ ft. lb. Thus THE EXTERNAL MECHANICAL WORK, IN FT. LB., DONE BY A GAS KEPT AT CONSTANT PRESSURE IS EQUAL TO THE PRESSURE (LB. PER SQ. FT.) MULTIPLIED BY THE INCREASE IN VOLUME (CU. FT.).

Suppose the initial and final temperatures to have been T_1 and T_2. Then by Charles' law

$$\frac{V_2}{V_1} = \frac{T_2}{T_1}, \quad \frac{V_2}{V_1} - 1 = \frac{T_2}{T_1} - 1, \quad \frac{V_2 - V_1}{V_1} = \frac{T_2 - T_1}{T_1}$$

and

$$V_2 - V_1 = \frac{V_1}{T_1}(T_2 - T_1).$$

Hence

$$W = P_1(V_2 - V_1) = \frac{P_1V_1}{T_1}(T_2 - T_1).$$

But $P_1V_1 = wRT_1$, hence

$$\frac{P_1V_1}{T_1} = wR.$$

Then $W = wR(T_2 - T_1)$. Then

$$H = T + I + \frac{W}{778} = T + I + \frac{wR(T_2 - T_1)}{778}$$

which we may put equal to $kw(T_2 - T_1)$ since the process is one of constant pressure. Note that T_2 and T_1 are TEMPERATURES, while T is the heat consumed in changing temperature, as in Art. 2.

For heating at CONSTANT VOLUME, on the other hand, $W = 0$ and $H = T + I = lw(T_2 - T_1)$. This expresses the value of $T + I$ for ANY process from a state 1 to a state 2. The value of $T + I$ depends on the two states only, and not on what happened in moving from one to the other of them. See Art. 18. Applying this value of $T + I$ to a process at CONSTANT PRESSURE,

$$kw(T_2 - T_1) = lw(T_2 - T_1) + \frac{wR}{778}(T_2 - T_1)$$

or

$$k - l = \frac{R}{778}. \tag{14}$$

Thus the quantity R in the perfect gas equation is simply THE DIFFERENCE OF THE TWO SPECIFIC HEATS expressed in ft. lb. It is the external work done in heating 1 lb. of gas 1° at constant pressure.

Prob. 44. Given k for hydrogen = 3.409, find l.

(Ans., 2.424 B.t.u.)

Prob. 45. What value of R is indicated for air, with k and l as given? (Ans., 53.37.)

Prob. 46. 1 lb. of air at 69° F. is placed in a vertical cylinder. A piston without weight rests above this air. There is no friction, so that the pressure on the air is that of the standard atmosphere above, or $14.696 \times 144 = 2116.2$ lb. per sq. ft.

The cylinder cross-sectional area is 1 sq. ft. State the pressure, volume and temperature of the air in the cylinder, using standard units. (Ans., $P = 2116.2$, $V = 13.34$, $T = 529$.) What will be the volume after heating this air 1°? The mechanical work done during such heating? (Ans., 13.365 cu. ft.: 53 ft. lb.)

17. **Ratio of Specific Heats.** The RATIO OF k TO l, called y, is of very great importance in Thermodynamics. Its value for air is $0.2375 \div 0.1689 = 1.406$. For all of the permanent gases, its value is close to 1.4. The following relations should be noted:

$$k - l = \frac{R}{778}, \quad \frac{k}{l} = y, \quad l = \frac{R}{778(y-1)},$$

$$k = \frac{Ry}{778(y-1)}, \quad y = \frac{778k}{778k - R}, \quad R = 778l(y-1).$$

Prob. 47. Find y for hydrogen. (Ans. 1.406.)

Prob. 48. If for nitrogen $k = 0.2438$, make two estimates of the probable value of l. (Ans., taking $R = 55.1$, $l = 0.173$. Taking $y = 1.4$, $l = 0.174$.)

18. **Joule's Law.** It has been demonstrated experimentally that with perfect gases the amount of disgregation work is so small that it may be ignored. FOR THE PERMANENT GASES, UNDER NORMAL CONDITIONS AND WITH SUFFICIENT ACCURACY FOR MOST ENGINEERING APPLICATIONS, $I = 0$ and $H = T + W$. This refers to Equation (3). In a PERFECT gas, I would be absolutely zero (Joule's law).

It follows from Joule's law that $H = T$ during a constant volume (zero work) operation. A constant volume process may therefore be employed to measure the T effect per unit of heat supplied. It is found that H is l B.t.u. per lb. per degree of temperature change in such a process. Then for any process whatever, $H = lw(T_2 - T_1) + W$, where W is expressed in B.t.u. This principle underlies much of what follows. For a given weight of a given gas, the W effect depends on the process and may range from zero upward: but the T effect is determined solely by the temperature change, no matter what the process is or how the change is effected.

Prob. 49. The temperature of 8 lb. of air is raised 125° while mechanical work equivalent to 100 B.t.u. is done by the air. How much heat is supplied? (Ans., 268.9 B.t.u.)

Prob. 50. By what type of process can the temperature of a gas be increased at the least expenditure of heat? (Constant volume process.) What is this expenditure for 100 lb. of hydrogen heated 10°? (Ans., 2424 B.t.u.)

Prob. 51. In an air compressor, 778 000 ft. lb. of work are done ON the air, which meanwhile rises 1000° in temperature. If 5 lb. of air are present, state values of T, I, W and H.

(Ans., $T = +844.5$, $I = 0$, $W = -1000$, $H = -155.5$.)

Thermodynamic Properties of Permanent Gases

Gas	Symbol	Boiling Point at Atmospheric Pressure, Deg. Fahr.	Molecular Weight	R	k	l	y
Helium	He	−450	4.0	386	1.25	0.75	1.667
Hydrogen	H_2	−412	2.016	766	3.409	2.424	1.406
Nitrogen	N_2	−318	28.02	55.1	0.2438	0.173	1.409
Oxygen	O_2	−297	32	48.25	0.2175	0.1555	1.399
Air	—	−314	—	53.36	0.2375	0.1689	1.406
Methane	CH_4	−265	16.032	96.31	0.5929	0.4698	1.26
Carbon monoxide	CO	−313	28	55.14	0.2425	0.1716	1.413

Note: $R = \frac{1544}{m}$ and $l = k - \frac{R}{778}$.

19. **Graphical Representation of External Work.** It has been shown that the external work done at constant pressure is equal to the pressure multiplied by the increase of volume. In Fig. 6, let *ab* represent a constant-pressure process. Horizontal distances represent volumes. Vertical distances represent pressures. Then $W = P_{ab}(V_b - V_a)$ = the AREA *abdc*. If the volume DECREASES, as when the process is FROM *b* TO *a*, the work is represented by the same area, but it is work done ON the gas, and has a NEGATIVE sign.

In Fig. 6, consider ANY path, *ef*. Take a short portion, *gh*, of this path. When *gh* is infinitely short, it may be considered horizontal. Then $W_{gh} = ghkj$. The work done along the whole

path *ef* is the sum of such areas as *ghkj*, or is the area *efml*. If the path is from left to right (*e* to *f*), *W* is +; otherwise, it is −. The following theorem may be stated:

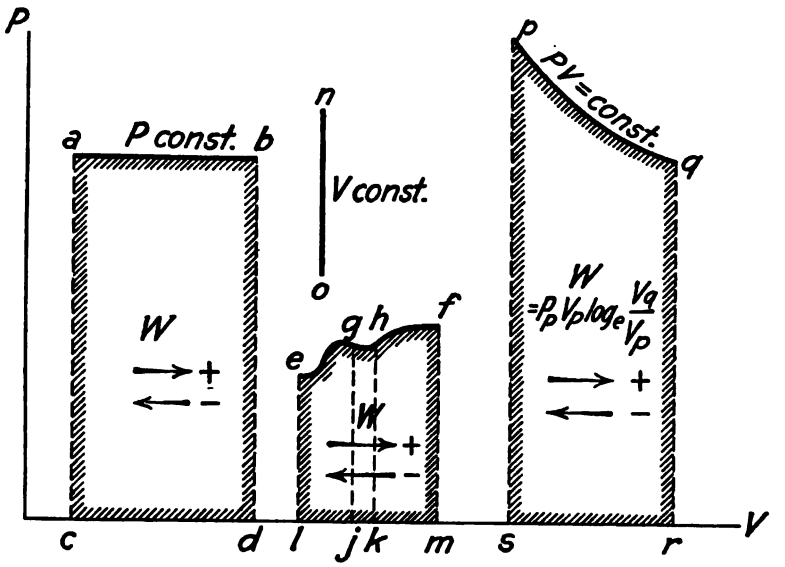

FIG. 6. Areas Representing External Work.

ON A *PV* DIAGRAM, THE VALUE OF *W* FOR ANY PROCESS IS REPRESENTED BY THE AREA BELOW THE LINE REPRESENTING THAT PROCESS.

For a constant-volume process, *no*, Fig. 6, there is no subtended area and hence $W = 0$, as already inferred.

20. **Isothermal Process.** An isothermal operation is one in which THE TEMPERATURE REMAINS CONSTANT. The gas must then conform to Boyle's law and the process is represented (Fig. 6) by an equilateral hyperbola *pq*. The area below this curve, *pqrs*, represents *W*. The magnitude of this area can be obtained by integration, the equation of the curve being $P_pV_p = P_qV_q$. It is

$$W = P_pV_p \log_e \frac{V_q}{V_p} \text{ ft. lb.}, \tag{15}$$

where $\log_e = 2.303 \log_{10}$. The symbol log (without a subscript) means $\log_{10}$.

In Fig. 5*a*, *psxv*, *ab*, *mlyn*, are isothermals.

For an isothermal process, the fundamental equation, $H = T + I + W$, becomes $H = W$: for $T = 0$ for an isothermal and $I = 0$ for a permanent gas. To obtain H in B.t.u., divide W by 778.

Prob. 52. If 10 lb. of air at standard atmospheric pressure and 32° F. are heated at constant pressure to 524° F., find H and W. (Ans., $H = 1168.5$ B.t.u., $W = 262\,134$ ft. lb.)

Prob. 53. In the previous problem, show that numerically $H - (W/778) = lw(T_2 - T_1)$.

Prob. 54. If 10 lb. of air at standard atmospheric pressure and 32° F. are expanded isothermally to twice the original volume, find W and H. (Ans., $W = 181\,730$ ft. lb.: $H = 233.6$ B.t.u.)

Prob. 55. If 10 lb. of air are cooled at constant volume from 524° to 32°, find W and H. (Ans., $W = 0, H = -830.99$ B.t.u.)

21. **Cycle.** Most thermodynamic operations consist of a series of processes which restore the fluid to its original condition. Such a series constitutes a CYCLE, which is represented by a closed figure on a PV diagram. For an entire cycle, the fundamental equation $H = T + I + W$ becomes $H = W$: because $T = 0$ when the original temperature is regained and $I = 0$ for a permanent gas. Hence,

IN A CLOSED CYCLE, THE NET ABSORPTION OF HEAT EQUALS THE NET EXTERNAL WORK DONE.

ΣH and ΣW must be expressed in the same units, either heat units or foot-pounds. This refers to a clockwise cycle. If the cycle is counter-clockwise, the law holds, but ΣH and ΣW are both negative. The area of the cycle itself is the algebraic sum of subtended areas under the paths which compose it: hence the area of the cycle represents ΣH and ΣW.

THE AREA OF A CLOSED CYCLE ON A PV DIAGRAM REPRESENTS THE NET AMOUNT OF HEAT ABSORBED OR

EMITTED AND ALSO THE NET AMOUNT OF EXTERNAL WORK DONE BY OR ON THE SUBSTANCE: both in ft. lb.

Prob. 56. In the cycle *abc* of Fig. 7, for 10 lb. of air, find the external work done along each path and the heat absorbed along each path and show that $\Sigma H = \Sigma W$. See Probs. 52–55. (Ans., $H_{ab} = 1168.5$, $H_{bc} = -830.99$, $H_{ca} = -233.6$, $\Sigma H = 103.9$ B.t.u. = 80 400 ft. lb.: $W_{ab} = 262\,134$, $W_{bc} = 0$, $W_{ca} = -181\,730$, $\Sigma W = 80\,400$ ft. lb.)

22. **Efficiency of Cycle.** The object of a thermodynamic operation is the performance of some net amount of external work. This is the result of the RECEPTION of heat. Since EFFICIENCY = EFFECT ÷ CAUSE,

THE EFFICIENCY OF A CYCLE IS THE NET AMOUNT OF HEAT ABSORBED OR EXTERNAL WORK DONE (AREA IN B.T.U.), DIVIDED BY THE WHOLE AMOUNT OF HEAT RECEIVED.

In other words, it is ΣH divided by the sum of all of the H terms having + signs. In Fig. 7, it is *abc* in B.t.u. divided by H_{ab}.

Prob. 57. Find the efficiency in Fig. 7. (Ans., 104 ÷ 1168.5 = 0.089.)

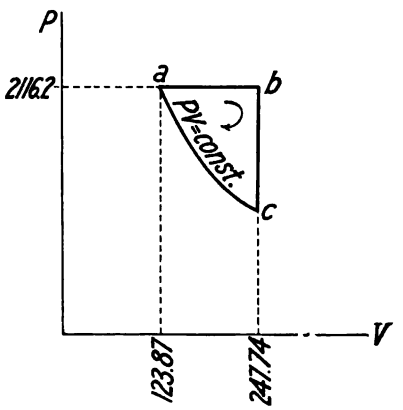

FIG. 7. Problem.

23. **Clearance, p_m, Displacement.** If such a cycle as that of Fig. 7 were actually employed, the DISPLACEMENT (PER 10 LB. OF AIR USED) would be $V_c - V_a = 123.87$ cu. ft. The CLEARANCE (Art. 7) would be $V_a \div (V_c - V_a) = 1.0$, or 100 per cent. The MEAN EFFECTIVE PRESSURE (Art. 7) would be $p_m = abc \div (V_c - V_a) = (104 \times 778) \div 123.87 = 653$ lb. per sq. ft. or 4.54 lb. per sq. in. If 100 lb. of air were used per min., the engine would develop (100/10) × (80 400/33 000) = 24.5 hp. (see Prob. 56).

Prob. 58. In the cycle of Fig. 8, find clearance, p_m, displacement and hp. if 50 lb. of air are used per min. and the cycle represents 1 lb. of air. State the absolute temperature at each point. (Ans., clearance = 0.333: displacement = 26.68 cu. ft. per lb. of air or 1334 cu. ft. per min., p_m = 20.83 lb. per sq. in., hp. = 121.273, T_d = 500°, T_a = 1000°, T_b = 4000°, T_c = 2000°.)

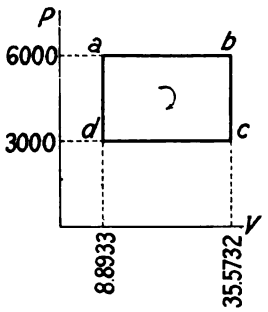

FIG. 8. Problem.

24. **General Case of Gas Process: the Polytropic.** Practically all heat engine processes may be represented on the PV diagram by smooth curves, V increasing as P decreases. Fig. 9 shows several such curves. An equation which represents them is PV^n = const. (the exponent applying to V only). When several such curves (called POLYTROPIC curves) pass through a given point, like a, the value of n indicates the STEEPNESS of the curve. Equality of n values does not imply coincident curves. For $n = 1$, for example, one curve is plotted through a and another through b.

For $n = 0$, PV^0 = const., or P = const. — the process is one of CONSTANT PRESSURE.

For $n = 1$, PV = const. — the process is ISOTHERMAL.

For $n = \infty$, $PV^\infty = \infty$, $P^{1/\infty}V$ = const., V = const. — the operation is at CONSTANT VOLUME.

25. **Relations of Properties in a Polytropic Process.** Consider a case of polytropic (PV^n = const.) expansion from 1 to 2. Then $P_1V_1^n = P_2V_2^n$. The substance being a gas and the weight constant, the perfect gas equation may be written $P_1V_1T_2 = P_2V_2T_1$. Dividing the second of these equations by the first,

$$\frac{T_2}{T_1} = \left(\frac{V_2}{V_1}\right)^{1-n}. \tag{16}$$

Extracting the n root of the first equation, and then dividing the second, by it, and transposing,

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{(n-1)/n}. \tag{17}$$

When $n = 0$, Equation (16) reduces to Charles' law for constant pressure. When $n = \infty$, Equation (17) reduces to Charles' law for constant volume.

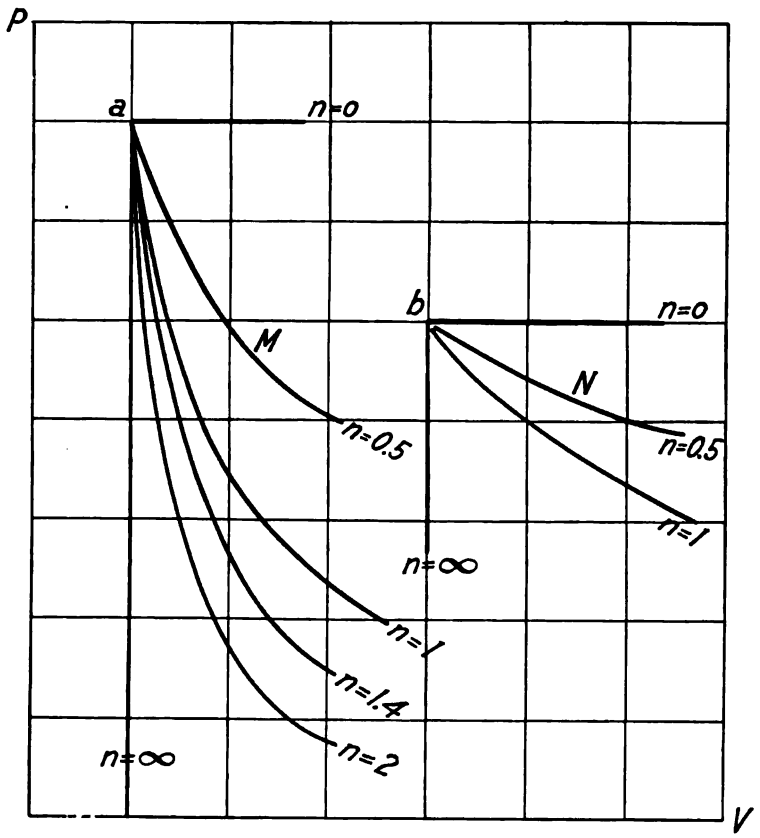

FIG. 9. Polytropic Processes.

Thus suppose $T_1 = 492$, $P_1 = 2116.2$, $V_1 = 12.387$. Let $n = 1.4$, $V_2 = 123.87$. Then $T_2 = 492 \times 10^{-0.4}$, $\log T_2 = 2.692 - 0.4 = 2.292$, $T_2 = 195.9$. Also (since $P_2/P_1 = (V_1/V_2)^n$), $P_2 = 2116.2 \times (1/10)^{1.4}$, $\log P_2 = 3.3256 - 1.4 = 1.9256$, P_2

$= 84.26$. As a check, from $P_1V_1T_2 = P_2V_2T_1$, $T_2 = (84.26 \times 123.87 \times 492) \div (12.387 \times 2116.2) = 195.9$.

Prob. 59. Given $P_1 = 10\,000$, $V_1 = 10$, $V_2 = 50$, $n = 2$. The substance is 4 lb. of nitrogen. Find T_1, T_2, P_2.

(Ans., $T_1 = 453.72$, $T_2 = 90.74$, $P_2 = 400$.)

26. **External Work, Polytropic Process.** The area under a curve passing through a high-pressure point a and a low-pressure point b, if the equation of the curve is $P_aV_a^n = P_bV_b^n$, is by integration,

$$W = \frac{P_aV_a - P_bV_b}{n - 1}. \tag{18}$$

This gives the work in ft. lb., which is to be regarded as + if the process is FROM a TO b (left to right). If the reverse, W is −. For $n = 1$, the work is indeterminate by this method, and Equation (15) should be used.

Prob. 60. In the problem immediately preceding, how much external work is done? (Ans., $W = +\ 80\,000$ ft. lb.)

27. **Heat Absorbed, Polytropic.** In any gas process, $\boldsymbol{H} = \boldsymbol{T} + \boldsymbol{W}$. For all processes, $\boldsymbol{T} = lw(T_2 - T_1)$. For a polytropic,

$$\boldsymbol{W} = \frac{P_1V_1 - P_2V_2}{n - 1} \text{ ft. lb.}$$

From Equation (11), $P_1V_1 = wRT_1$ and $P_2V_2 = wRT_2$. Hence

$$\boldsymbol{W} = \frac{wR}{778(n - 1)}(T_1 - T_2) \text{ B.t.u.}$$

Hence for a polytropic,

$$\boldsymbol{H} = \boldsymbol{T} + \boldsymbol{W} = w(T_2 - T_1)\left(l - \frac{R}{778(n - 1)}\right).$$

Noting that

$$\frac{R}{778} = k - l, \qquad \boldsymbol{H} = w(T_2 - T_1)\left(l - \frac{k - l}{n - 1}\right),$$

and putting $k = ly$,

$$\boldsymbol{H} = w\left(l\frac{n - y}{n - 1}\right)(T_2 - T_1). \tag{19}$$

But for any process, if the specific heat is s, $\boldsymbol{H} = sw(T_2 - T_1)$. Hence for a polytropic, $s = l(n - y)/(n - 1)$. The value of s for a given gas depends on n alone. The polytropic is then a curve of CONSTANT SPECIFIC HEAT. This is why the polytropic is so important. Practically all permanent gas processes are represented by polytropic curves. So also are many vapor processes. Many practical applications will be found in Chapter III. For $n = 0, s = ly = k$, as is to be expected. If $n > 1 < y$, s is NEGATIVE, i.e., heat is EMITTED when the temperature RISES or ABSORBED when the temperature FALLS. This actually occurs in every air compressor. During compression, the temperature of the air rises although the jacket water is continually abstracting heat from the air. The reason is that we are doing external work on the air faster than we are absorbing heat from it. For POSITIVE values of s, a rise of temperature indicates reception of heat. When $n = 1$, $s = \infty$; a condition necessary for an isothermal process, in which no finite supply of heat can ever change the temperature.

Prob. 61. In Prob. 59, what is the specific heat? (Ans., 0.1022.) How much heat is absorbed or emitted? (Ans., $\boldsymbol{H} = -$ 148 B.t.u.) What is the value of $\boldsymbol{T}$ in $\boldsymbol{H} = \boldsymbol{T} + \boldsymbol{I} + \boldsymbol{W}$? (Ans., $-$ 251 B.t.u.)

28. **Adiabatic.** An adiabatic process is one during which NO HEAT IS RECEIVED OR EMITTED by the fluid in question. Under adiabatic conditions, $\boldsymbol{H} = \boldsymbol{T} + \boldsymbol{I} + \boldsymbol{W} = 0$: but $\boldsymbol{T}$ and $\boldsymbol{W}$ are both finite. (If the substance is a permanent gas, $\boldsymbol{I} = 0$.) If we assume that the adiabatic may be represented by an equation in the form $PV^n = \text{const.}$, n being undetermined as yet,

$$\boldsymbol{W} = -\boldsymbol{T} \qquad \text{or} \qquad \frac{P_1V_1 - P_2V_2}{778\,(n-1)} = lw(T_1 - T_2),$$

where the process is from state 1 to state 2. Since $P_1V_1 = wRT_1$, $P_2V_2 = wRT_2$, this leads to $l = R/778(n - 1)$. In Art. 17, it was shown that $l = R/778(y - 1)$. Hence FOR AN ADIABATIC PROCESS $n = y$. The equation of the adiabatic is $P_1V_1^y = P_2V_2^y$ and for such process, following Arts. 25, 26,

$$\frac{T_2}{T_1} = \left(\frac{V_2}{V_1}\right)^{1-y} = \left(\frac{P_2}{P_1}\right)^{(y-1)/y},$$

$$W = \frac{P_1V_1 - P_2V_2}{y - 1}.$$

During an adiabatic expansion, the temperature falls. In Fig. 5*a*, *ab* is an isothermal, *ae* an adiabatic. The two curves projected to the PV plane appear as *zk*, *zg*.

Prob. 62. Air expands adiabatically from $P_1 = 10\,000$, $V_1 = 10$ to $V_2 = 30$. Find P_2 and W_{12}.

(Ans., $P_2 = 2140$, $W_{12} = 88\,100$ ft. lb.)

29. **Summary.** The following table summarizes what has been learned concerning permanent gas processes. The values of H are in B.t.u.: those of W are in ft. lb. if P is taken in lb.

Permanent Gases—Processes

Name	Constant Property	H	W	s	n	Remarks
Horizontal...	Pressure	$kw(T_2 - T_1)$	$P_{12}(V_2 - V_1)$	k	0	$V_2/V_1 = T_2/T_1$
Vertical.....	Volume	$lw(T_2 - T_1)$	0	l	∞	$P_2/P_1 = T_2/T_1$
Isothermal...	Temperature	$\frac{P_1V_1 \log_e \frac{V_2}{V_1}}{778}$	$P_1V_1 \log_e \frac{V_2}{V_1}$	∞	1	$P_1V_1 = P_2V_2$
Adiabatic....	Entropy*	0	$\frac{P_1V_1 - P_2V_2}{y - 1}$	0	y	$\frac{T_2}{T_1} = \left(\frac{V_2}{V_1}\right)^{1-y} = \left(\frac{P_2}{P_1}\right)^{\frac{y-1}{y}}$
Polytropic....	Specific heat	$sw(T_2 - T_1)$	$\frac{P_1V_1 - P_2V_2}{n - 1}$	$l\,\frac{n-y}{n-1}$	...	$\frac{T_2}{T_1} = \left(\frac{V_2}{V_1}\right)^{1-n} = \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}$

* The meaning of this term may be disregarded for the present.

per sq. ft. Only the last line of the table need be emphasized. With the exceptions ***in heavy type***, all of the other necessary relations may be derived from those of the polytropic by substituting the proper value for n.

30. **Application.** A cycle will now be worked out in detail to illustrate these formulas and to show how a complete check of

the computations is possible. In Fig. 10, data are as follows:

Clockwise cycle, 2 lb. of nitrogen

$P_1 = P_2 = 140\,000,$

$P_2V_2^{\nu} = P_3V_3^{\nu},$

$V_1 = V_5 = 0.5,$

$T_2 = 1270.32,$

$P_5 = 80\,000,$

$P_5V_5 = P_4V_4,$

$P_4 = 20\,000, \quad V_3 = 3,$

$P_4V_4^{n} = P_3V_3^{n}$, with n unknown.

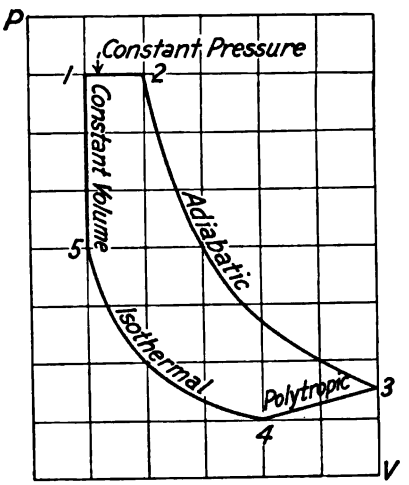

FIG. 10. Cycle for Computation.

The unknown properties are first found:

$$T_1 = \frac{P_1V_1}{wR} = \frac{140\,000 \times 0.5}{2 \times 55.1} = 635.16,$$

$$T_5 = T_1\frac{P_5}{P_1} = 635 \times \frac{80\,000}{140\,000} = 363 = T_4,$$

$$V_2 = V_1\frac{T_2}{T_1} = 0.5 \times \frac{1270}{635} = 1.0,$$

$$V_4 = V_5\frac{P_5}{P_4} = 0.5 \times \frac{80\,000}{20\,000} = 2.0,$$

$$P_3 = P_2\left(\frac{V_2}{V_3}\right)^{1.409} = 140\,000\left(\frac{1}{3}\right)^{1.409},$$

$$T_3 = T_2\left(\frac{V_3}{V_2}\right)^{-0.409} = 1270 \times 3^{-0.409},$$

$$\log P_3 = 5.1461 - 0.6723 = 4.4738, \quad P_3 = 29\,770,$$

$$\log T_3 = 3.1040 - 0.1951 = 2.9089, \quad T_3 = 810.5.$$

Also, to check,

$$T_3 = \frac{P_3V_3}{wR} = \frac{29\,770 \times 3}{2 \times 55.1} = 810.5.$$

To find the value of n for the process $4 - 3$:

$$P_4V_4^n = P_3V_3^n, \quad \log P_4 + n \log V_4 = \log P_3 + n \log V_3,$$

$$n = \frac{\log P_3 - \log P_4}{\log V_4 - \log V_3},$$

$$n = \frac{4.4738 - 4.3010}{0.3010 - 0.4771} = -0.981.$$

The negative sign is unusual: note the slope of the curve. It does not affect the method of procedure. The specific heat for the path $4 - 3$ is

$$s_{43} = l\frac{n - y}{n - 1} = 0.173 \times \frac{-2.390}{-1.981} = +0.2087.$$

We now compute the heat quantities:

$$H_{12} = wk(T_2 - T_1) = 2 \times 0.2438(1270 - 635) = 309.7 \quad (+)$$

$$H_{23} = 0$$

$$H_{34} = ws(T_4 - T_3) = 2 \times 0.2087(363 - 810.5) = 186.8 \quad (-)$$

$$H_{45} = \frac{P_5V_5 \log_e \frac{V_4}{V_5}}{778} = \frac{80\,000 \times 0.5 \times 2.3 \log 4}{778} = 71.2 \quad (-)$$

$$H_{51} = wl(T_1 - T_5) = 2 \times 0.173(635 - 363) = 94.1 \quad (+)$$

$$\Sigma H = 146 \text{ B.t.u.}$$

The external work quantities are:

$$W_{12} = P_{12}(V_2 - V_1)$$

$$= 140\,000(1.0 - 0.5) = 70\,000 \quad (+)$$

$$W_{23} = \frac{P_2V_2 - P_3V_3}{y - 1}$$

$$= \frac{(140\,000 \times 1.0) - (29\,770 \times 3)}{0.409} = 123\,900 \quad (+)$$

$$W_{34} = \frac{P_3V_3 - P_4V_4}{n-1}$$

$$= \frac{(29\,770 \times 3) - (20\,000 \times 2)}{-1.981} = 24\,900 \quad (-)$$

$$W_{45} = P_5V_5 \log_e \frac{V_4}{V_5}$$

$$= 80\,000 \times 0.5 \times 2.3 \log 4 = 55\,400 \quad (-)$$

$$W_{51} = 0$$

$$\Sigma W = 113\,600 \text{ ft. lb.}$$

$$= 146 \text{ B.t.u.}$$

Thus $\Sigma W = \Sigma H$.

31. **Isodiabatics** are polytropic curves not coincident, but having the same value of n. In Fig. 9, the lines M and N are isodiabatics. The following theorems will be found later to save much time in computation with many common forms of cycle:

1. Let a pair of isodiabatics, M and N, Fig. 11, be cut by lines of constant PRESSURE, P_1 and P_2. Then

$$P_eV_e^n = P_gV_g^n; \quad P_fV_f^n = P_hV_h^n.$$

Dividing the first of these equations by the second,

$$\frac{P_eV_e^n}{P_fV_f^n} = \frac{P_gV_g^n}{P_hV_h^n} \quad \text{or since } P_e = P_f \text{ and } P_g = P_h,$$

$$\frac{V_e}{V_f} = \frac{V_g}{V_h} = \frac{T_e}{T_f} = \frac{T_g}{T_h}, \tag{20}$$

the equality of temperature ratios arising from Charles' law.

2. Let a pair of isodiabatics, M and N, be cut by lines of constant VOLUME, V_1 and V_2. Then

$$P_jV_j^n = P_lV_l^n, \quad P_kV_k^n = P_mV_m^n,$$

$$\frac{P_jV_j^n}{P_kV_k^n} = \frac{P_lV_l^n}{P_mV_m^n} \quad \text{or} \quad \frac{P_j}{P_k} = \frac{P_l}{P_m} = \frac{T_j}{T_k} = \frac{T_l}{T_m}, \tag{21}$$

Charles' law being again applied.

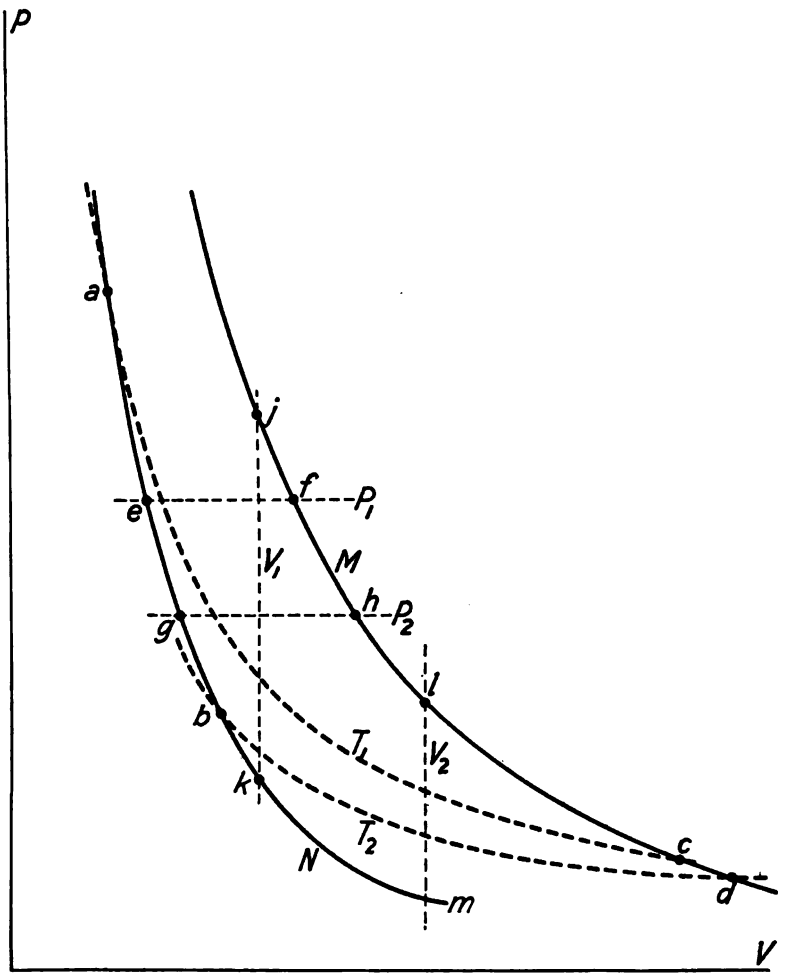

FIG. 11. Isodiabatics.

3. Let a pair of isodiabatics, M and N, be cut by lines of constant TEMPERATURE, T_1 and T_2. Then

$$\frac{T_a}{T_b} = \left(\frac{P_a}{P_b}\right)^{(n-1)/n} = \frac{T_c}{T_d} = \left(\frac{P_c}{P_d}\right)^{(n-1)/n},$$

$$\frac{P_a}{P_b} = \frac{P_c}{P_d}, \quad \frac{P_a}{P_c} = \frac{P_b}{P_d} = \frac{V_c}{V_a} = \frac{V_d}{V_b}, \tag{22}$$

Boyle's law being applied in this instance.

Nearly all heat engines work in cycles made up of one or more pairs of isodiabatics (see, for example, Art. 38).

Prob. 63. In Fig. 11, given $t_k = 40°$ F., $t_j = 1040°$ F., $P_k = 20$, $P_l = 27.3$, find P_j and P_m. (Ans., $P_j = 60$, $P_m = 9.1$.)

Prob. 64. In Fig. 11, if $V_e = 15$, $V_f = 30$, $t_f = 904°$ F., $V_h = 36$, $t_g = 260°$ F., find V_g, T_h and T_e.

(Ans., $V_g = 18$, $T_h = 1440$, $T_e = 682$.)

Prob. 65. In Fig. 11, $P_d = 10$, $P_b = 26.5$, $V_d = 74$, $V_c = 69$, $T_c = 750$, $T_d = 575$. Find V_b, P_c, P_a, V_a.

(Ans., $V_b = 27.9$, $P_c = 14$, $P_a = 37.1$, $V_a = 26$.)

32. **Imperfect Gases: Application to a Gun.** An imperfect gas may be conceived as a perfect gas through which there are distributed incompressible particles. Let the volume of such particles in 1 lb. of gas be a cu. ft. Then the remainder of the gas conforms to Equation (11), or

$$P(V - wa) = wRT. \tag{23}$$

This is the Van der Waals equation for imperfect gases. The gas liberated in the powder chamber of a gun furnishes a good example of an imperfect gas. For powder gases, the value of a is approximately $0.001v_0$, where v_0 = specific volume at 32° F. and normal barometer. Hence

$$a = 0.001v_0 = \frac{0.001 \times 492R}{14.696 \times 144} = 0.000232R.$$

Prob. 66. Find a when $R = 46$. (Ans., 0.0107.)

Prob. 67. The powder chamber of a gun has a volume of 8.5 cu. ft. The weight of powder is 300 lb., the temperature 1000° F. before the projectile moves. Assuming R for powder gases = 46, find the powder pressure at the moment the projectile begins to move. Compute first by Equation (11) and afterward by Equation (23). (Ans., $p = 16\,400$ if the gas is perfect: $p = 26\,500$ by the Van der Waals equation.)

Prob. 68. If the gun in the foregoing problems is 12-in., 50 ft. long from seat to muzzle, what is the whole volume of gas behind the projectile when it emerges from the muzzle?

(Ans., 47.77 cu. ft.)

Prob. 69. If maximum pressure occurs when the projectile has traveled half way from seat to muzzle, what is the volume of gases behind the projectile then? (Ans., 28.13 cu. ft.) If maximum pressure occurs when the projectile has moved 19.6 ft., what is the volume at that point? (Ans. 23.88 cu. ft.)

Prob. 70. In the second case of the foregoing problem, assume that a plot of pressure against volume would be a straight line from atmospheric pressure and 8.5 cu. ft. volume to 30 000 lb. per sq. in. pressure at 23.88 cu. ft. volume. How much external work is done up to the point of maximum pressure?

(Ans., 33 200 000 ft. lb.)

Prob. 71. In the foregoing, the further expansion of the powder gases is polytropic, with $n = 7/6$. What is the pressure at the muzzle? (Ans., 13 400 lb. per sq. in.) How much external work is done from point of maximum pressure to muzzle? (Ans., 66 000 000 ft. lb.)

Prob. 72. In all of the foregoing (Probs. 70, 71) if the powder contains 1620 B.t.u. per lb., what is the thermodynamic efficiency of the gun? (Ans., 0.26.)

Prob. 73. Ignoring frictional and other losses, the total mechanical work becomes $\frac{1}{2}MU^2$ where M = mass of projectile (= lbs. weight ÷ 32.17) and U = muzzle velocity, ft. per sec. Find muzzle velocity if the projectile weighs 800 lb.

(Ans., 2820 ft.)

Prob. 74. In the preceding problems, how long would the gun need to be in order to reduce the muzzle pressure to that of the atmosphere? (Ans., about 4 miles.)

Prob. 75. Under the conditions adopted above, find the temperature (by the Van der Waals equation) at the point of maximum pressure and at the muzzle. (Ans., 6500°: 6200°: abs.)

33. **Graphical Representation of Heat Absorbed.** In Fig. 12, let *ab* represent any permanent gas process. Draw two adiabatics through *a* and *b*. At an indefinite distance to the right the two adiabatics will meet. Designate the meeting-point as *M*. Then the figure *abM* is a cycle and the area *abM*, as already shown, represents the net amount of heat absorbed or rejected in that cycle. There are only three processes included in the cycle: *Ma*, *ab* and *bM*. We know by definition of an adiabatic that $H_{Ma} = 0$, $H_{bM} = 0$. Hence H_{ab} alone is repre sented by the area *abM*: or

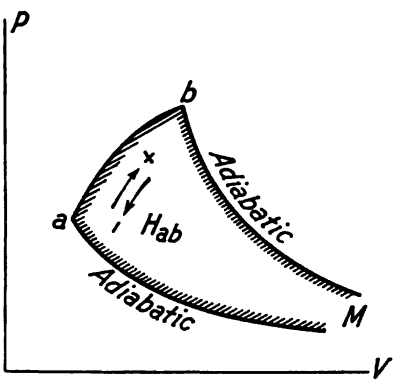

FIG. 12. Heat Absorbed, Any Process.

THE HEAT ABSORBED OR EMITTED ALONG ANY PATH IS REPRESENTED BY THE AREA INCLUDED BETWEEN THAT PATH AND TWO ADIABATICS PASSING THROUGH ITS EXTREMITIES AND PROLONGED INDEFINITELY TO THE RIGHT.

If, as viewed from a point lying between the adiabatics, the path is from left to right, *H* is +. Otherwise, *H* is −. Along an adiabatic, by this theorem, $H = 0$.

34. **Carnot Cycle.** In Fig. 13, a cycle *abdc* is formed by the combination of a pair of isothermals, *ab* and *cd*, with a pair of adiabatics, *ac* and *bd*. As in all cycles, the efficiency is NET HEAT ÷ GROSS HEAT. The net heat is the area *abdc*. The "gross" or plus heat is made up of four possible terms: but of these $H_{ac} = H_{bd} = 0$, the paths in question being adiabatic: while H_{dc} is eliminated because for a clockwise cycle its value is negative. Hence the efficiency is

$$e = \frac{abdc}{mabM} = \frac{H_{ab} - H_{dc}}{H_{ab}}$$

$$= \frac{P_a V_a \log_e \frac{V_b}{V_a} - P_c V_c \log_e \frac{V_d}{V_c}}{P_a V_a \log_e \frac{V_b}{V_a}}.$$

From Equation (22), $V_b/V_a = V_d/V_c$: hence

$$e = \frac{P_a V_a - P_c V_c}{P_a V_a} = \frac{T_a - T_c}{T_a} = \frac{T_1 - T_2}{T_1}, \qquad (24)$$

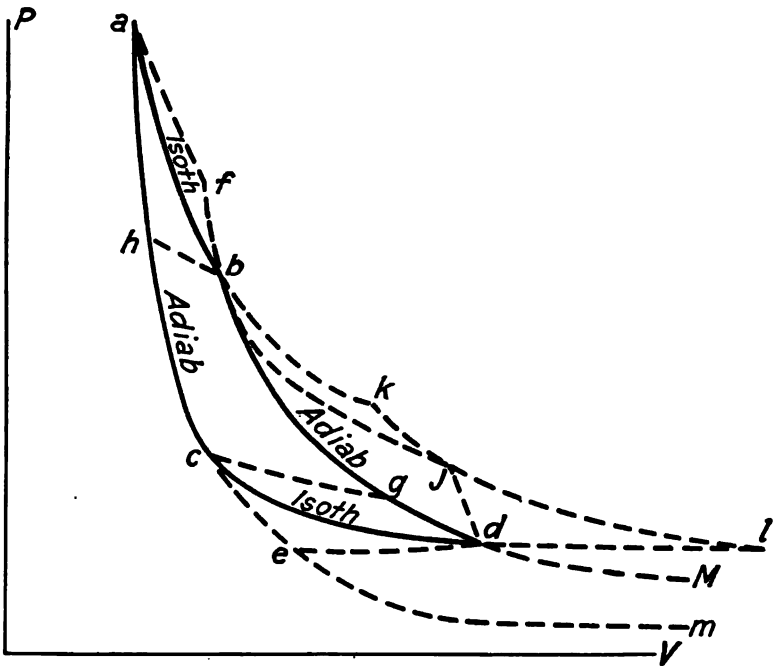

FIG. 13. Carnot Cycle.

where T_1 and T_2 are the highest and lowest (absolute) temperatures of the cycle.

THIS VALUE IS REALIZED IN A CARNOT CYCLE, NO MATTER WHAT GAS IS USED.

BETWEEN ASSIGNED LIMITING TEMPERATURES, NO POSSIBLE CYCLIC PROCESS CAN HAVE AN EFFICIENCY EXCEEDING THAT OF THE CARNOT CYCLE. In Fig. 13, if the proposed superior cycle

were *abde*, the point *e* would be below the lower limiting temperature. If it were *afdc*, the point *f* would lie above the upper limiting temperature. Neither condition is allowable under the stipulation made. An allowable modified cycle would be *abgc*. As compared with the Carnot, this gives a reduced amount of external work represented by the area *cgd*: but the gross heat absorbed is still *mabM* and the efficiency is hence reduced. Consider next the cycle *hbdc*. It gives less work than the Carnot, represented by the area *abh*: but it saves heat, the gross heat being now *mhbM*. The saving of heat is exactly equal to the loss of work. Then the efficiency of the new cycle is

$$\frac{abdc - abh}{mabM - abh}.$$

Since $abdc/mabM < 1.0$, the efficiency of the new cycle is less than that of the Carnot cycle.

If some such form as *abjdc* were proposed, it would be surpassed in efficiency by a broadened-out Carnot cycle *aklc*, and hence by the original cycle *abdc*, which lies between the same temperature limits as *aklc*.

NO HEAT ENGINE CAN HAVE AN EFFICIENCY EXCEEDING $(T_1 - T_2)/T_1$ WHERE T_1 AND T_2 ARE THE HIGHEST AND LOWEST ABSOLUTE TEMPERATURES OF THE FLUID IN THE ENGINE.

Equally definite, but lower, limits of efficiency will presently be established for various specific types of engine.

Prob. 76. A Carnot cycle gives, area = 100, gross heat = 200. A competitive cycle *hbdc* is formed, with *abh* = 20. State the efficiency of each cycle. (Ans., 0.50 and 0.444.)

Prob. 77. What is the least amount of coal (12 725 B.t.u. per lb.) per hour per horse power used for an engine using steam at 340° F. and exhausting to a condenser at 140° F.?

(Ans., 0.8 lb.)

Prob. 78. The maximum temperature in an oil engine is 3500° abs., the exhaust temperature 1000° abs. The engine develops 190 hp. and uses oil of 20 000 B.t.u. per lb. What is the least possible oil consumption per hr.? (Ans., 33.8 lb.)

35. **Carnot Cycle Reversed.** In Fig. 13, suppose the cycle to be worked counter-clockwise. Begin at *c*. Suppose the gas to be very cold—say at 0° F. Let it pass through a room, receiving heat (area *mcdM*) but not increasing in temperature. The point *d* is reached. Then compress the gas adiabatically to *b*. Its temperature will rise—say to 150° F. Then pass it across coils through which cold water is circulating. Heat will be abstracted (area *mabM*) but the temperature may be kept constant by adjusting the rates of flow of air and water. THE HEAT EMITTED ALONG *ba* IS THE HEAT THAT WAS ABSORBED ALONG *cd* (plus something). Then expand the gas adiabatically along *ac*. the temperature falling and the initial condition being restored,

We have now done work ON the gas, or EXPENDED POWER. For what purpose? In order that heat might be taken up by the gas at 0° and discharged by it at 150°. What good object does this serve?

Suppose it is desired to make ice. Water is available at 50°. Our gas, at 0°, can cool and freeze this water. For the operation to continue, it must then get rid of the heat which it has taken from the water. By expending power we raise the gas temperature WITHOUT ADDING HEAT and the heat taken from the water frozen can then be transferred to the circulating water in the coils of the "condenser" and so discharged from the system.

This is the operation of a REFRIGERATING MACHINE. In the Carnot cycle used for refrigeration, all changes of temperature would take place without any flow of heat, and all transfers of heat would take place without changes of temperature. Actual refrigerating machines do not realize these conditions.

This is unfortunate. The efficiency of the Carnot cycle used for refrigeration is

$$e = \frac{mcdM}{mabM - mcdM} = \frac{T_2}{T_1 - T_2}. \tag{25}$$

A reversal of the argument of Art. 34 shows that this is the maximum possible value. No actual refrigerating machine can surpass this efficiency.

Efficiency is effect divided by cause. The effect, in refrigera-

tion, is the absorption of heat by the fluid at low temperature. The cause is the mechanical work done. The part played by the circulating water being disregarded, as in Equation (25), the efficiency may exceed 100 per cent. It is a maximum when the temperature range is least. With ENGINES, on the other hand, efficiency is a maximum when the temperature range is greatest.

Prob. 79. What would be the percentage saving of coal, in Prob. 77, if the exhaust temperature could be reduced to 40° F.? (Ans., $33\frac{1}{3}$ per cent.)

Prob. 80. With the gas temperatures specified in Art. 35, what is the efficiency of refrigeration in the Carnot cycle? (Ans., 3.07 or 307 per cent.) What would it be if heat were rejected at 50° F.? (Ans., 920 per cent.)

Prob. 81. Under the gas temperatures specified in Art. 35, the heat absorbed by the gas (useful refrigeration) is 307 000 B.t.u. per hr. What amount of mechanical work must be done on the gas? (Ans., 100 000 B.t.u. per hr.) What engine horse power is ideally required? (Ans., 39.3.)

36. **Availability of Heat.** Recapitulating: for efficient power generation in a Carnot cycle, the heat should be supplied at a high temperature and discharged at a low temperature. The latter temperature should be low, but is more or less fixed by surrounding conditions. The supply temperature is the one which is controllable. It should be high for high efficiency of the Carnot cycle. No actual engine can exceed the efficiency of the Carnot cycle. Anything that keeps the latter efficiency low is bound to keep the former efficiency low, also. Hence for all engines, a high supply temperature is desirable. Anything that lowers temperature lowers the potential efficiency. A fall of temperature implies a loss of possible power. The availability of heat is determined by its temperature. A foot pound is 1/778 of a heat unit: but a much smaller fraction of a heat unit is actually realized in mechanical work by actual

engines. Low fluid supply temperatures make the fraction particularly small. Steam at 600° F. is worth potentially about twice as much as steam at 300° F. It does not cost twice as much.

Whenever the temperature falls without the simultaneous performance of mechanical work there is a loss of availability of the heat which can never be recovered. This principle is sometimes referred to as the SECOND LAW of Thermodynamics.

37. **Carnot Cycle Impracticable.** An engine should be efficient and powerful. The latter condition renders it necessary that the mean effective pressure, p_m, should be fairly high. This should be accomplished without excessive maximum pressures. In the Carnot cycle, Fig. 13,

$$\frac{P_b}{P_d} = \left(\frac{T_1}{T_2}\right)^{y/(y-1)} = \left(\frac{T_1}{T_2}\right)^{3.5},$$

nearly, since for any perfect gas y is approximately 1.4. Thus if T_1 is twice T_2 (which would give an efficiency of 0.50), P_b would be $11.3P_d$. P_a is still greater. Suppose we narrow the Carnot cycle by moving *ac* toward *bd*. Ultimately the two curves coincide, the enclosed area becomes zero, $p_m = 0$ and still the maximum pressure is 11.3 times the lowest pressure for the temperature range stated.

For a finite area, the mean effective pressure is

$$p_m = \frac{\text{area}}{V_d - V_a} = \frac{\Sigma W}{V_d - V_a} = \left(p_a V_a \log_e \frac{V_b}{V_a} + \frac{p_b V_b - p_d V_d}{y-1} - p_c V_c \log_e \frac{V_d}{V_c} - \frac{p_a V_a - p_c V_c}{y-1} \right) \div (V_d - V_a)$$

$$= \left\{ (p_a V_a - p_c V_c) \log_e \frac{p_a}{p_b} + \frac{p_b V_b - p_a V_a}{y-1} + \frac{p_c V_c - p_d V_d}{y-1} \right\} \div (V_d - V_a).$$

Since $p_c V_c = p_d V_d$ and $p_b V_b = p_a V_a$, the last two terms within the bracket disappear. Then

$$p_m = \frac{(T_a - T_c)wR \log_e \frac{p_a}{p_b}}{wR\left(\frac{T_d}{p_d} - \frac{T_a}{p_a}\right)} = \frac{(T_1 - T_2)\log_e\left\{\frac{p_a}{p_d}\left(\frac{T_2}{T_1}\right)^{y/(y-1)}\right\}}{\frac{T_2}{p_d} - \frac{T_1}{p_a}}. \quad (26)$$

For $T_2/T_1 = 0.5$, $y = 1.4$, $p_d = 15$, $p_a = 300$, $T_1 = 1000$, this gives

$$p_m = \frac{500 \times 2.3 \log (20 \times 0.5^{3.5})}{\frac{500}{15} - \frac{1000}{300}} = 9.5 \text{ lb. per sq. in.}$$

Thus, although the efficiency is 50 per cent., the mean effective pressure is only 9½ lb. per sq. in., in spite of the fact that the maximum pressure is 300 lb. per sq. in. This would never do. It would require a very large cylinder in order to develop even a small amount of power. A STEAM engine, not working in the Carnot cycle, might easily give $p_m = 100$ when the maximum pressure was 300 lb. It would thus be more than ten times as powerful as the Carnot engine. This would compensate for its lower efficiency.

Prob. 82. A Carnot cycle has an efficiency of 0.4. Find the lowest possible value of P_a/P_d, using $y = 1.4$. (Ans., 5.98.) What is then the value of p_m? (Ans., 0.)

CHAPTER III

AIR ENGINES, AIR COMPRESSORS, AIR REFRIGERATION

Ideal Cycle

38. **Joule Cycle.** A cycle underlying the operation of several types of air machinery is shown in Fig. 14, diagram *abcd*. It is bounded by a pair of adiabatics and a pair of constant-pressure lines (hence by two pairs of isodiabatics — Art. 31). The EFFICIENCY is $\Sigma H/H_{ab}$, H_{dc} being negative and H_{da} and H_{bc} being both zero (compare Art. 34).

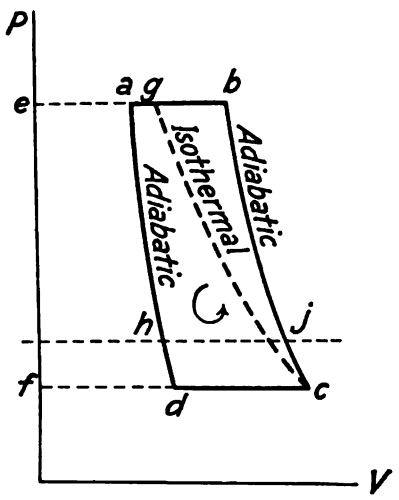

FIG. 14. Joule Cycle.

$$e = \frac{H_{ab} - H_{cd}}{H_{ab}} = 1 - \frac{H_{cd}}{H_{ab}} = 1 - \frac{kw(T_c - T_d)}{kw(T_b - T_a)},$$

where w = weight of fluid and k = specific heat at constant pressure. From Equation (20),

$$\frac{T_c}{T_d} = \frac{T_b}{T_a}, \quad \frac{T_c}{T_d} - 1 = \frac{T_b}{T_a} - 1, \quad \frac{T_c - T_d}{T_b - T_a} = \frac{T_d}{T_a},$$

$$e = 1 - \frac{T_d}{T_a} = \frac{T_a - T_d}{T_a}. \qquad (27)$$

This looks like the expression for efficiency of the Carnot cycle (Art. 34), which is, however, in terms of the LIMITING temperatures. For Fig. 14, the Carnot efficiency is $(T_b - T_d)/T_b$, which is HIGHER than the Joule efficiency.

The COMPRESSION RATIO (ratio of pressures during the adiabatic compression *da* which closes the cycle) being called C, we have

$$C = \frac{P_a}{P_d} = \left(\frac{T_a}{T_d}\right)^{y/(y-1)}, \quad e = 1 - C^{(1-y)/y}. \tag{28}$$

Prob. 83. Find the efficiency of a Joule cycle with a compression ratio of 10 when $y = 1.406$. (Ans., 0.486.) If the lowest temperature is 500° abs., and $y = 1.406$ what is the temperature at the end of adiabatic compression? (Ans., 972° abs.)

Prob. 84. If T_b, Fig. 14, is 1500° abs., what is the Carnot efficiency in the foregoing problem? (Ans., 0.667.)

39. **Mean Effective Pressure.** The exponent y of the curves *bc* and *ad* is simply a special value of the polytropic exponent n. The mean effective pressure (average ordinate) is

$$p_m = \frac{abcd}{V_c - V_a}$$

$$= \frac{p_a(V_b - V_a) + \dfrac{p_bV_b - p_cV_c}{n-1} - p_c(V_c - V_d) - \dfrac{p_aV_a - p_dV_d}{n-1}}{V_c - V_a}$$

(Arts. 16, 26)

$$= \frac{n}{n-1}(p_bV_b - p_aV_a - p_cV_c + p_dV_d) \div (V_c - V_a)$$

$$= \frac{n}{n-1}\{p_b(V_b - V_a) + p_d(V_d - V_c)\} \div (V_c - V_a).$$

Now CLEARANCE $= c = V_a \div (V_c - V_a)$: hence $(V_c - V_a) = V_a/c$. Also $p_b/p_d = C$.

$$p_m = \frac{cnp_d}{n-1} \cdot \frac{C(V_b - V_a) + V_d - V_c}{V_a}$$

$$= \frac{cnp_d}{n-1}\left(C\frac{V_b}{V_a} - C + \frac{V_d}{V_a} - \frac{V_c}{V_a}\right).$$

Substitute

$$\frac{V_d}{V_a} = C^{1/n}, \quad \frac{V_c}{V_a} = \frac{c+1}{c}, \quad \frac{V_b}{V_a} = \frac{V_b}{V_c} \cdot \frac{V_c}{V_a} = C^{-1/n} \cdot \frac{c+1}{c}.$$

$$p_m = \frac{cnp_d}{n-1}\left(C^{(n-1)/n} \cdot \frac{c+1}{c} - C + C^{1/n} - \frac{c+1}{c} \right)$$

$$= \frac{np_d}{n-1}\{(c+1)(C^{(n-1)/n} - 1) + c(C^{1/n} - C)\}. \qquad (29)$$

This holds whether $n = y$ or not.

Prob. 85. In Fig. 14, p_d = normal atmospheric pressure, $C = 10$, $T_d = 500$, $T_b = 1500$, $n = 1.4$. Find T_a (Ans., 965), T_c (Ans., 778), V_c/V_d (Ans., 1.56), V_d/V_a (Ans., 5.18), V_c/V_a (Ans., 8.05), c (Ans., 0.142), p_m (Ans., 19.6).

Compressed Air

40. **Air Machinery.** This section deals only with those air machines in which the pressure changes are of sufficient magnitude to introduce noticeable temperature effects. It excludes such devices as centrifugal fans and blowers. It considers the air as pure and dry and treats the fluid as a perfect gas. Properties of moist air are to be considered later.

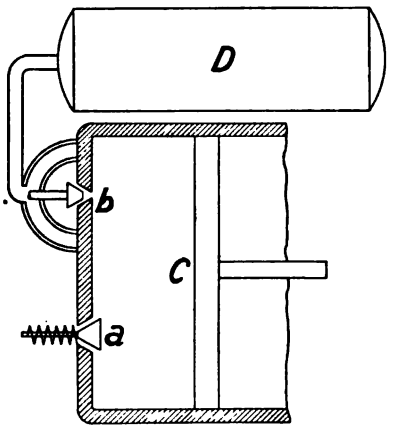

FIG. 15. Air Compressor.

41. **Compressor.** A compressor is the reverse of an engine. Work is done ON the fluid (air) by the supply of power from a steam cylinder, an electric motor, or some other suitable source. Fig. 15 represents the essential parts of a compressor cylinder C, with piston, inlet valve a, discharge valve b and receiver D. Suppose the piston to be at the extreme left, and both valves to be closed. It is moved toward

the right. At a suitable point early in this stroke, the inlet valve is caused to open. The piston suction draws air into the cylinder throughout the rest of the stroke. At the end of the stroke, the inlet valve is closed. The piston now moves to the left, compressing the air. When the pressure has been sufficiently increased, the discharge valve opens. The high-pressure air is then discharged to the receiver and pipe line throughout the remainder of this left-hand stroke. When the stroke is completed, the discharge valve is closed. The air left in the clearance space expands, during the first part of the succeeding right-hand stroke. When its pressure has been reduced to that of the atmosphere, the inlet valve opens and the cycle is repeated.

Practically all compressors are double-acting.

In Fig. 14, the inlet valve opens at *d*. Air is drawn in along *dc*. The pressure along *dc* is a pound or so below that of the atmosphere. The inlet valve closes at *c* and the air is compressed as along *cb*. If the compression were infinitely RAPID, the compression path would be an adiabatic, for there would be no time for heat to escape. If compression were infinitely SLOW, the curve would be an isothermal, because any increase of temperture would be offset by conduction of heat. Power is saved in the latter case. The lower the value of n the less is the area of the cycle (which is COUNTER-CLOCKWISE) and the less is the power consumed. The value of n depends on the piston speed and the thoroughness of cooling by the water jacket which surrounds the cylinder. (Some small compressors are air-cooled only.) Usually, n is between 1.3 and 1.35. The discharge valve opens at *b*. The pressure along *ba* is a pound or so higher than the receiver pressure and does not vary much, because of the relatively large receiving capacity into which the air is discharged. The discharge valve closes at *a*. The clearance air then expands as along *ad*, the value of n for this curve being practically the same as for *cb*.

42. **Compressor Power Consumption and Capacity.** Equation (29) gives a very close approximation to the mean effective

pressure of an actual compressor if a proper value is selected for n. The value of p_d may be taken in good machines at 14 lb. per sq. in. Clearance averages 0.04. C is determined by the discharge pressure desired. If S = piston speed, ft. per min., N = r.p.m., d = piston diameter, in., D = displacement, cu. ft. per min., the indicated horse power of the compressor cylinder (compare Art. 7) is (double-acting)

$$\text{Ihp.} = \frac{\pi}{4} \cdot \frac{p_m d^2 S}{33\,000} = \frac{144 p_m D}{33\,000}$$

and the stroke in ft. is $S \div 2N$. Values of S are from 400 to 800. The primary horse power required to drive the compressor is the above power divided by the mechanical efficiency of the compressor and its driver.

The IDEAL action might be considered as realized when there was no clearance, when $p_d = 14.696$ lb. per sq. in., and when compression (path cg) was at constant temperature. Then

$$D = V_c - V_e = V_c,$$

$$\Sigma W = W_{eg} + W_{gc} - W_{cf}$$

$$= P_g(V_g - V_e) + P_g V_g \log_e \frac{V_c}{V_g} - P_c(V_c - V_f).$$

But $V_e = 0$, $V_f = 0$, $P_g V_g = P_c V_c$.

$$\Sigma W = P_c V_c \log_e \frac{P_g}{P_c} = 144 \times 14.696 \times 2.3026 D \log\left(\frac{p_g}{14.696}\right),$$

$$\text{Ihp.} = \frac{\Sigma W}{33\,000} = 0.148 D\,(\log p_g - 1.167), \tag{30}$$

$$p_m = \frac{\Sigma W}{144 D} = 33.84\,(\log p_g - 1.167). \tag{31}$$

Here (since D is taken as displacement PER MINUTE) Fig. 14 is to be regarded as a summation of all cycles realized in one minute of time.

In the ideal cycle, the volume of air drawn into the cylinder at atmospheric pressure is $V_c = D$. In an actual cycle the volume is always less than D. Suppose in the cycle $abcd$ the

line hj to represent normal atmospheric pressure. The whole volume of air compressed, measured at this pressure, is V_j. Part of this is clearance air; the volume of which, also measured at atmospheric pressure, is V_h. Then the volume DRAWN IN AND COMPRESSED, measured at atmospheric pressure, is $V_j - V_h$.

FREE AIR is air at normal atmospheric pressure.

COMPRESSOR CAPACITY is measured by the volume of free air drawn in and compressed per minute (symbol F).

VOLUMETRIC EFFICIENCY (not an efficiency in the thermodynamic sense) is the ratio of free air drawn in and compressed. to the displacement.

$$e_v = F/D. \tag{32}$$

For the ideal cycle $fcge$, Fig. 14, $F = D$ and $e_v = 1.0$. For the actual cycle $dcba$,

$$e_v = \frac{F}{D} = \frac{V_j - V_h}{V_c - V_a} = (c+1)\left(\frac{p_c}{14.696}\right)^{1/n} - c\left(\frac{p_a}{14.696}\right)^{1/n}: \tag{33}$$

for,

$$V_c - V_a = \frac{V_a}{c}, \quad V_j = V_c\left(\frac{p_c}{14.696}\right)^{1/n} = V_a \cdot \frac{c+1}{c}\left(\frac{p_c}{14.696}\right)^{1/n},$$

$$V_h = V_a\left(\frac{p_a}{14.696}\right)^{1/n}.$$

High volumetric efficiency (which leads to large capacity for a given size and speed), results when the clearance is low, the valve ports and passages of ample area, the piston speed low and the value of n low. It is usually between 0.70 and 0.90.

Prob. 86. An air compressor of 2000 cu. ft. capacity has an intake pressure of 13.226 lb. per sq. in., absolute, 4 per cent. clearance, a discharge pressure of 132.26 lb. gauge. Find p_m if $n = 1.35$. (Ans., 35.2 lb.)

Prob. 87. In the previous problem, what is the volumetric efficiency? (Ans., 0.742.)

Prob. 88. In the foregoing, find displacement and Ihp. (Ans., 2700 cu. ft. per min., 415 hp.)

Prob. 89. The compressor cylinder is directly driven by a steam cylinder and the steam cylinder develops 532 Ihp. What is the mechanical efficiency? (Ans., 0.78.)

Prob. 90. What would be the displacement of an ideal compressor giving the diagram *fcge*, Fig. 14? (Ans., 2000 cu. ft. per min.) What would be its Ihp.? (Ans., 296.)

Prob. 91. In the actual compressor (Prob. 88), the piston speed is 800 ft. per min. and the rotative speed 100 r.p.m. Find cylinder diameter and piston stroke.

(Ans., diameter = 24.9 in., stroke = 4 ft.)

43. **Cooling.** Low values of n for the compression curve are realized as a result of jacket cooling. The limit of gain arises from the slow rate of heat transfer from air through the cast iron walls of the cylinder (Chapter VII). Below this limit, the heat lost by the air, in B.t.u. per min. is WEIGHT × SPECIFIC HEAT × TEMPERATURE CHANGE

$$w_a s_a (T_c - T_b)$$

$$= \left\{ \frac{P_c(1+c)D}{53.36\,T_c} \right\} \left\{ 0.1689 \frac{n - 1.406}{n - 1} \right\} \left\{ T_c - T_c \left(\frac{p_b}{p_c} \right)^{(n-1)/n} \right\}.$$

This may be equated to the heat gained by the water

$$= w_w(t_2 - t_1),$$

in all of which w_a and w_w are weights of air compressed and water circulated, lb. per min., s_a = specific heat of air for a polytropic path having the exponent n, t_2 and t_1 are outlet and inlet temperatures of water at the jacket. Normal values are $T_c = 660$ (it cancels out anyway), $t_2 = 130$, $t_1 = 60$ or 70.

Prob. 92. In the problems of Art. 42, find s_a, w_a and w_w if $T_c = 660$, $t_1 = 70°$, $t_2 = 130°$. (Ans., $s_a = -0.027$, $w_a = 152$, $w_w = 39$.) What is the temperature at b? (Ans., 767° F.)

44. **Two-Stage Compressor.** Compression in successive stages saves power. In Fig. 16, compression begins at c. ct is an adiabatic, cz a polytropic, cs an isothermal. The last is the

preferred process, but cannot be realized because heat can be transmitted only slowly through the cylinder walls.

Compress from c to u adiabatically, or slightly better than that if possible, then discharge the air to an external cooler which reduces its temperature to v. Again compress and cool (vw, wx). With an infinite number of steps and complete cooling at each

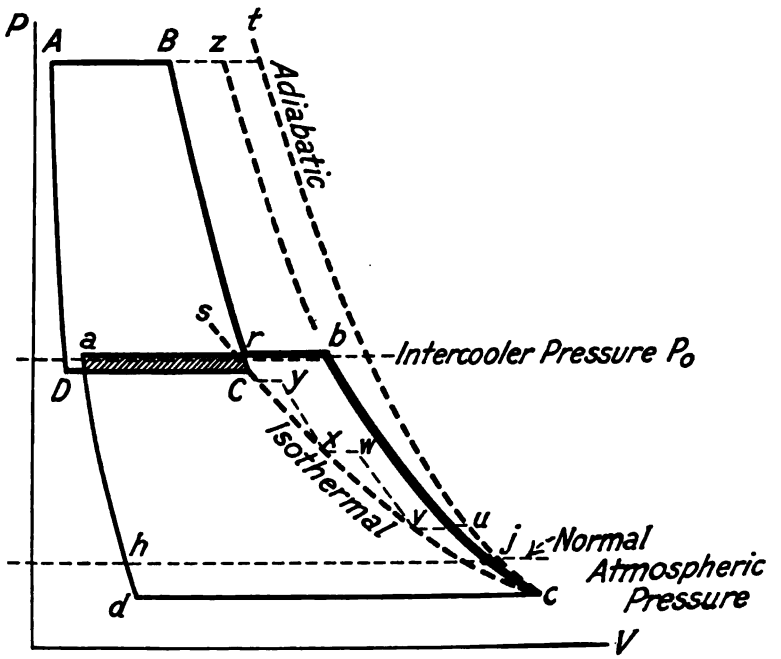

FIG. 16. Two-Stage Compressor.

step, the isothermal path is realized, and power is saved. A number of cylinders and intercoolers are required for multi-stage compression. The intercooler is a cylindrical shell filled with small tubes through which cold water circulates, while the air surrounds the tubes, on its way from one cylinder to the next.

The diagrams *abcd* and *ABCD* show the action of a two-stage machine, with polytropic compression in each cylinder. With good intercooling, $t_C = t_c$. The gross saving of work, as com-

pared with single stage compression, is represented by the area *Bzbr*. To offset this, partially, there is half the duplicated hatched area due to friction of air passing through the intercooler, which makes the suction pressure of the second cylinder a pound or so less than the discharge pressure of the first, and thus somewhat decreases the saving.

Multi-stage operation saves power, though increasing first cost and complication. The saving is greatest if the discharge pressure is high. The higher this pressure, the greater the number of stages desirable. Two stages are common in shop practice, with discharge pressures around 100 lb. gauge.

45. **Power Required in Two-Stage Compression.** The least total power is consumed when the intercooler pressure is

$$P_0 = \sqrt{P_B P_c}.$$

The two cylinders will then (if $t_C = t_c$) require equal amounts of power. The suction pressure p_C, with properly designed valve passages, may be taken at $p_0 - 0.5$, and the discharge pressure, p_a, at $p_0 + 0.5$. Equation (29) gives the mean effective pressure of either cylinder, but the following points should be observed:

First (low-pressure) cylinder, clearance $= c_l = v_a/(v_c - v_a)$, $C = p_a/p_d$.

Second (high-pressure) cylinder, clearance $= c_h = v_A/(v_C - v_A)$, $C = p_A/p_D$, p_d should read p_D. The equations of Art. 42 may then be used to obtain the Ihp. of each cylinder, for a given displacement, D, of the LOW-pressure cylinder. With perfect intercooling, as shown in Fig. 16, the high-pressure displacement is $D \times p_c/p_C$.

Thus, take the conditions of Prob. 86.

$$p_0 = \sqrt{13.226 \times 146.96} = 44.1.$$

Then $p_C = 43.6$, $p_a = 44.6$. For the first cylinder, the mean effective pressure is

$$p_{ml} = \frac{13.226n}{n-1}\left[(c_l+1)\left\{\left(\frac{44.6}{13.2}\right)^{(n-1)/n} - 1\right\} + c_l\left\{\left(\frac{44.6}{13.2}\right)^{1/n} - \left(\frac{44.6}{13.2}\right)\right\}\right].$$

For the second cylinder, it is

$$p_{mh} = \frac{43.6n}{n-1}\left[(c_h+1)\left\{\left(\frac{147.0}{43.6}\right)^{(n-1)/n}-1\right\} + c_h\left\{\left(\frac{147.0}{43.6}\right)^{1/n}-\left(\frac{147.0}{43.6}\right)\right\}\right].$$

The displacement of the second cylinder is

$$D_H = D_L \times \frac{13.2}{43.6}.$$

The horse power consumed in the two cylinders is

$$\frac{144D_l}{33\,000}\left(p_{ml}+\frac{13.2}{43.6}p_{mh}\right).$$

46. **Capacity, Two-Stage.** The volumetric efficiency e_{vl} of the first cylinder is given by Equation (33) and $D_l = F \div e_{vl}$. It is higher than the volumetric efficiency of the single-stage machine with the same EXTREME pressure limits (p_A and p_c) and the same clearance.

Prob. 93. In Art. 45, with $n = 1.35$, $c_l = c_h = 0.04$, find p_{ml} and p_{mh}. (Ans., $p_{ml} = 18$, $p_{mh} = 58$.)

Prob. 94. In the foregoing, what is the ratio of displacements of the two cylinders? (Ans., $D_L/D_H = 3.3$.)

Prob. 95. In the foregoing, what is the volumetric efficiency of the first cylinder? Compare Prob. 87. (Ans., 0.87.)

Prob. 96. If $F = 2000$, $E_{vl} = 0.87$, find D_L, D_H, in the foregoing. (Ans., $D_L = 2300$, $D_H = 700$.)

Prob. 97. Find the horse power consumed by the whole machine, noting that the cylinders require practically equal powers. (Ans., 356 hp.) How much power is saved over single stage operation as computed in Prob. 88? (Ans., 59 hp.)

Prob. 98. With $S = 800$, r.p.m. $= 100$, state cylinder dimensions. (Ans., both strokes are 4 ft.: diameters are 23 and 12.65 in.)

47. **Heat Removed in Two-Stage Compression.** The method of calculation is much the same as for single-stage machines, given in Art. 43. In Art. 45, and Probs. 93–98, assume $T_c = 660$. Then $T_b = T_c(p_b/p_c)^{(n-1)/n} = 660(44.6/13.226)^{0.26} = 904$. Assume $T_C = 660$, as should easily be the case with reasonably good intercooling. Then $T_B = T_C(p_B/p_C)^{(n-1)/n} = 904$, also. The air temperature is reduced $904 - 660 = 244$ degrees in the intercooler. It is increased 244° in each cylinder. The weight of air in the low-pressure cylinder is $P_c(1 + c_l)D_l \div 53.36T_c = (13.226 \times 144 \times 1.04 \times 2300) \div (53.36 \times 660) = 130$ lb. per min. The weight in the second cylinder is practically the same if its clearance is the same. The weight in the intercooler may differ somewhat, since the intercooler does not cool the clearance air: this difference will be disregarded. The passage of air through the intercooler may be regarded as taking place at constant pressure. The specific heat there is consequently $k = 0.2375$. The specific heat of air during compression in the cylinders is

$$s = l\frac{n - y}{n - 1} = 0.1689 \times \frac{-0.056}{0.35} = -0.027.$$

Then the heat removed from the air is its weight multiplied by the sum of the products of specific heats and temperature ranges, or

$130\{(0.2375 \times 244) + (2 \times 0.027 \times 244)\} = 9250$ B.t.u. per min.

This must be removed by water supplied at the jackets and intercooler. It considerably exceeds the heat removed in single-stage compression, where there is no intercooler.

Prob. 99. How much heat was removed in Prob. 92?

(Ans., 2330 B.t.u. per min.)

Prob. 100. In Art. 47, 150.4 lb. of water are delivered to the intercooler every minute. How much will be the rise of temperature of this water? (Ans., 50°.)

Prob. 101. In Art. 47, the cooling water enters the jackets at 70° and leaves at 156.5°. How much water must be supplied at each jacket? (Ans., 9.9 lb. per min.)

48. **Multi-Stage Compression in General.** For four stages, the intercooler pressures are determined by an extension of the method of Art. 45. There will be three intercoolers. The pressure in the middle one, P_3, will be a mean proportional between the suction pressure, P_1, of the lowest pressure cylinder and the discharge pressure, P_5, of the highest pressure cylinder. The pressure in the first intercooler, P_2, will be $\sqrt{P_1P_3}$. That in the third will be $P_4 = \sqrt{P_3P_5}$. The method may be further extended for eight stages. For THREE stages, with successive suction pressures of P_1, P_2 and P_3 and a final discharge pressure of P_4, the condition for least power consumption and (with intercooling sufficient to restore the pre-compression temperature in each case) for equal power consumption by all three cylinders is

$$P_3 = \sqrt[3]{P_1P_4^2}, \qquad P_2 = \sqrt[3]{P_4P_1^2}.$$

Prob. 102. A four-stage compressor operates between pressures of 14 and 250 lb. absolute. What are the best intercooler pressures? (Ans., 29, 59, 122.)

Prob. 103. What would be the best intercooler pressures in a three-stage compressor for the same conditions?

(Ans., 37 and 96 lb.)

49. **Vacuum Pump.** The vacuum pump connected with a steam engine or turbine CONDENSER is an air compressor having a suction pressure between 0.5 and 2.0 lb. absolute and a discharge pressure around 15 or 16 lb. absolute. Because of the presence of water in the cylinder, the value of n is low.

50. **Air Engine.** The most common example of an engine using compressed air is the PNEUMATIC TOOL. Suppose some of the air compressed, in Prob. 86, to be used in an engine. The essential parts of an engine are shown in Fig. 17, the upper diagram indicating the cylinder, C; piston, P; piston-rod, R; inlet passage, I, from the compressor; and outlet passage, E, to the atmosphere. These passages have valves, not shown.

The piston is at the left-hand end of its stroke, position P_1. The inlet valve is open. Hence there is a pressure on the piston,

tending to force it toward the right. The compressor discharge pressure (Prob. 86) was 146.96 lb. absolute. The pressure in the engine cylinder will be somewhat less: we will take it at 145.53 lb., as indicated. In the lower diagram, the point a represents the state of the fluid, V_a being the clearance volume.

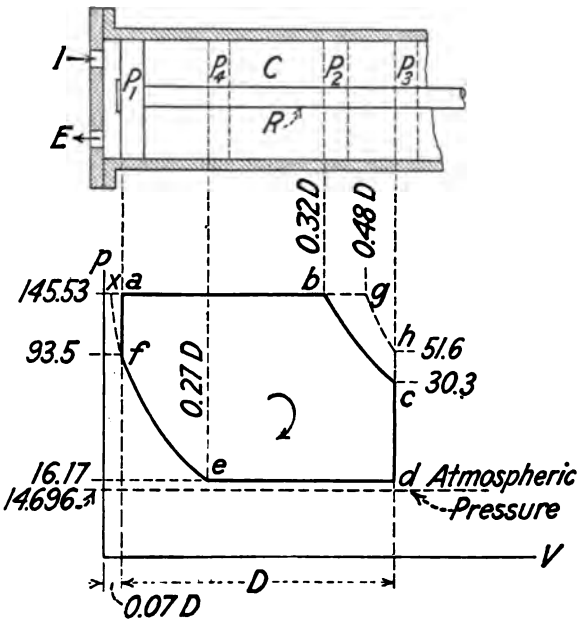

FIG. 17. Air Engine.

The piston moves toward the right, until the position P_2 is reached. The inlet valve is then closed (point b on the lower diagram). The piston will continue to move to the right, impelled by the expansive force of the air, giving the process curve bc. The equation of this curve is $p_b V_b^n = p_c V_c^n$, where n is usually around 1.3 to 1.35. High speed or a heat-insulated condition of the cylinder tend to increase the value of n and thus to decrease work and power. The temperature falls along bc.

Expansion terminates at c, when the discharge valve opens, air rushes out of the cylinder through the outlet E and the

pressure falls, the piston having reached the end of its stroke (path *cd*). The farther the expansion is carried, the more work will be obtained from the expansion of a given quantity of air. On the other hand, long expansion leads to low mean effective pressure and hence to large size of engine for a given power. We will assume CUT-OFF (the closing of the inlet valve at *b*) to occur at one-quarter stroke. (The diagram is not drawn to scale.)

Now on a volume scale, $V_d - V_a = D$, the displacement per stroke. If clearance be taken at 7 per cent., $V_a = 0.07D$. If cut-off is at quarter stroke, $ab = D/4$ or $0.25D$ and $V_b = 0.25D + 0.07D = 0.32D$. The pressure at c is $p_b(V_b/V_c)^n$. With $n = 1.3$, this gives $p_c = 145.53(0.32/1.07)^{1.3} = 30.3$.

The piston is now in the position P_3. It moves to the left (by the inertia of the reciprocating parts, or if the engine is double-acting, under the action also of high-pressure air on its right-hand side). Throughout the first part of the left-hand return stroke, the pressure remains about constant and a little above that of the atmosphere. (It must be somewhat above, else air would not flow out of the cylinder.) We will take the pressure at 16.17 lb.

At some position P_4, before the return stroke is completed, the outlet valve is closed. The comparatively small amount of air then remaining in the cylinder is compressed (path *ef*) during the remainder of the return stroke. The point *e*, at which the exhaust valve closes is called the point of COMPRESSION (strictly, "beginning" of compression). The object of compression is to cushion the action of the engine. We will assume compression to begin at 8/10 the return stroke. Hence $V_e = 0.27D$.

The path *ef* follows the law $p_f V_f^n = p_e V_e^n$, the value of n being again about 1.3. Then

$$p_f = p_e \left(\frac{V_e}{V_f}\right)^n = 16.17\left(\frac{0.27}{0.07}\right)^{1.3} = 93.5.$$

The compression curve terminates at *f*, the piston having then again reached its extreme left-hand position. The inlet valve immediately opens and the pressure increases, giving the path *fo* before the piston starts to move again.

51. **Air Engine Design.** Still referring to Fig. 17, the work done in the cylinder by the air is, since $W_{fa} = 0$ and $W_{cd} = 0$, $\Sigma W = W_{ab} + W_{bc} - W_{de} - W_{ef}$

$$= 144D\left\{ p_a(V_b - V_a) + \frac{p_bV_b - p_cV_c}{n-1} - p_d(V_d - V_e) - \frac{p_fV_f - p_eV_e}{n-1} \right\}$$

$$= 144D\left\{ (145.53 \times 0.25) + \frac{(145.53\times0.32)-(30.3\times1.07)}{0.3} - (16.17 \times 0.8) - \frac{(93.5 \times 0.07) - (16.17 \times 0.27)}{0.3} \right\}$$

$= 9120D$ ft. lb. per revolution, on each side of a double-acting piston.

Suppose 10 hp. to be required, the piston speed to be 300 ft. per min. and the piston diameter to equal the stroke. If N = r.p.m., S = piston speed, d = dia. in in., L = stroke in ft.,

$$9120D \times 2N = 10 \times 33\,000, \quad D = \frac{\pi}{4} \cdot \frac{d^3}{1728} = \frac{d^3}{2200},$$

$$N = \frac{S}{2L} = \frac{6S}{d}, \quad 9120 \times \frac{d^3}{2200} \times \frac{12S}{d} = 330\,000,$$

$$d = \sqrt{\frac{330\,000 \times 2200}{9120 \times 12 \times 300}} = 4.7 \text{ in.}, \quad L = 4.7 \div 12 = 0.392 \text{ ft.},$$

$$N = 1800 \div 4.7 = 383 \text{ r.p.m.}, \quad D = \frac{4.7^3}{2200} = 0.0472 \text{ cu. ft.}$$

52. **Air Consumption.** The volume of fresh air present at b, Fig. 17, when the supply is cut off, is $V_b = 0.32D$. Some of this is clearance air. How much? Not V_a, for some of the air at a is fresh air, admitted during the operation fa. We know that the volume of clearance air alone was $V_a = V_f$, AT THE PRESSURE p_f. What is it at the pressure p_a? It is reasonable to assume a continued compression of clearance air by incoming air, along the path fx, which is the path ef produced. Then the volume V_x is that of clearance air present at the pressure p_a, when the total volume of air received is V_b. The value of V_x is

$$V_e\left(\frac{p_e}{p_x}\right)^{1/n} = 0.27D\left(\frac{16.17}{145.53}\right)^{0.769} = 0.05D.$$

Hence the volume of fresh air supplied to the cylinder by the compressor is $V_b - V_x = 0.27D = 0.27 \times 0.0472 = 0.0127$ cu. ft. per stroke or $0.0127 \times 383 \times 2 = 9.7$ cu. ft. per min. This is air at 145.53 lb. pressure. Measured as FREE AIR, its volume would be $9.7 \times 145.53/14.696 = 96$ cu. ft. per min.

The AIR RATE, which measures the economy of the engine, is the VOLUME OF FREE AIR USED PER HOUR PER HP. Its value for our conditions is

$$\frac{96 \times 60}{10} = 576 \text{ cu. ft.}$$

53. **Comparison with Ideal Conditions.** The ideal air engine cycle would have no clearance, expansion would be "complete" (i.e., cut-off would occur so early that the points c and d would coincide), the exhaust pressure would be atmospheric and n would equal 1. Cycle *egcf*, Fig. 14, will represent the action if $p_{fe} = 14.696$. Then the fresh air = eg, free air = $eg \times p_g/p_e = fc$, and the work of the cycle is

$$P_e(V_g - V_e) + P_g V_g \log_e \frac{V_c}{V_g} - P_f(V_c - V_f)$$

$$= 144 \times 14.696F \times 2.3 \log \frac{p_g}{14.696} = 4860F \log \frac{p_g}{14.696},$$

F denoting the volume of free air used to produce this work. If s cycles are produced per hr., the free air consumption is Fs cu. ft. per hr. and the hp. is

$$\frac{4860Fs}{60 \times 33\,000} \log \frac{p_g}{14.696},$$

so that the IDEAL AIR RATE is

$$\frac{Fs}{\text{hp.}} = \frac{1\,980\,000}{4860\,(\log p_g - 1.167)} = \frac{407}{\log p_g - 1.167}. \qquad (34)$$

For our conditions, $p_g = 145.53$, $\log p_g = 2.163$, ideal air rate $= 407/0.996 = 409$ cu. ft. The "relative efficiency" of our engine, or efficiency as compared with that attainable under ideal conditions, is $409 \div 576 = 0.71$. (This is not a true efficiency, but only the RATIO of two efficiencies.)

The mean effective pressure of our actual engine is

$$p_m = \frac{W}{D} = \frac{9120}{144} = 63.3 \text{ lb. per sq. in.}$$

That of the ideal engine is given by Equation (31) and is for our conditions 33.6 lb. per sq. in. If both engines have the same dimensions and speed, the powers are in the same ratios as the mean effective pressures. The actual engine develops 10 hp. The ideal engine will therefore develop only

$$\frac{33.6}{63.3} \times 10 = 5.3 \text{ hp.}$$

Its lower mean effective pressure is due to its "complete" expansion, which makes the pV diagram depart more noticeably from a rectangle.

54. **Preheat.** Reference has been made to the fall of temperature which occurs along *bc*, Fig. 17. This is important. The air may cool below 32° F. Any moisture then present congeals and may obstruct the action of the engine. This condition may be avoided by using very little expansion, as in pneumatic tools. Another method is to heat the air, close to the engine, in a sort of hot air furnace or preheater. An interesting result follows the use of the preheater.

Assume in Fig. 17 that $T_b = 560°$ and that by employing a preheater this temperature can be raised to 840°. Then, without any change in the quantity of air admitted to the cylinder, its volume at cut-off becomes

$$V_g = V_b \frac{T_g}{T_b} = 0.32D \times \frac{840}{560} = 0.48D,$$

by Charles' law. The air now expands along *gh*, the law being such that

$$p_h = p_g \left(\frac{V_g}{V_h}\right)^n = 145.53 \left(\frac{0.48}{1.07}\right)^{1.3} = 51.6.$$

Added work is gained, equal to the area *bghc*, the value of which is $W_{bg} + W_{gh} - W_{bc}$

$$= P_g(V_g - V_b) + \frac{P_gV_g - P_hV_h}{n-1} - \frac{P_bV_b - P_cV_c}{n-1}$$

$$= 144D\left\{(145.53 \times 0.16) + \frac{(145.53 \times 0.48) - (51.6 \times 1.07)}{0.3} - \frac{(145.53 \times 0.32) - (30.3 \times 1.07)}{0.3}\right\}$$

$$= 144 \times 0.0472 \times 26.3 = 178 \text{ ft. lb. per cycle.}$$

This is equivalent (there being 766 cycles per min.) to 4.1 hp., ADDED TO THE ENGINE OUTPUT WITHOUT ANY INCREASE IN SIZE OR AIR CONSUMPTION.

What does this cost? The preheater must supply enough heat to warm all the air in the cylinder (both fresh air and clearance air) from 560° to 840° absolute. The specific heat may be taken as at constant pressure, $k = 0.2375$. The weight of air is $P_bV_b \div RT_b = (144 \times 145.53 \times 0.32 \times 0.0472) \div (53.36 \times 560) = 0.01056$ lb. per cycle. The preheat per cycle is then $0.01056 \times 0.2375 \times 280 = 0.7$ B.t.u. or 545 ft. lb., and the efficiency OF THE PREHEATING OPERATION is $178 \div 545 = 0.326$. The importance of this will now be shown.

55. **Efficiency of Compressed Air Plant.** In Probs. 86 and 88, it was found that a 2000 cu. ft. (free air capacity per minute) single-stage compressor discharging at 132.26 lb. gauge pressure (146.96 lb. absolute) required 415 indicated hp. in the air cylinder. Prob. 89 gives the corresponding hp. of the steam cylinder which drives the compressor as 532. Prob. 90 shows that the ideal compressor would have required only 296 hp., measured in its own cylinder.

If we start with 532 Ihp. steam, we lose at once $532 - 415 = 117$ hp., in friction of compressor mechanism. We then lose $415 - 296 = 119$ hp. in the compressor cylinder due to departure from ideal conditions. The "relative efficiency" or ratio of actual and ideal efficiencies of this cylinder is $296 \div 415 = 0.713$.

The compressed air reaches the engine (Art. 52) at a pressure somewhat less (145.53 lb. absolute) than that at which it was discharged from the compressor, and also at a much reduced

temperature. Having defined the ideal engine and compressor cycles as we have, the only loss chargeable to pipe line transmission is the pressure loss, which may be estimated as follows.

Art. 53 shows that the ideal air rate for our engine is 409 cu. ft. of free air per Ihp. hr. If we installed a sufficient number of engines to use all the air furnished by the compressor, the ideal Ihp. of all of these air engine cylinders would be $(2000 \times 60) \div 409 = 293$. The pipe line loss is then $296 - 293 = 3$ hp., or about 1 per cent. If the compressor discharge and engine inlet pressures had been the same, this loss would have been zero.

By Art. 52, the air rate of the actual engine is 576. Hence the total Ihp. available in actual engine cylinders is $(2000 \times 60) \div 576 = 208$. The "relative" efficiency of our engine is as already stated, $208 \div 293 = 0.71$.

The efficiency of the whole conversion process, from Ihp. of steam engine to Ihp. of air engine, is then $208 \div 532 = 0.391$. This is not a thermodynamic efficiency covering the conversion of heat to work, but merely an efficiency of conversion from work in one form to work in another form. A compressed air plant is merely a method for power distribution. A further loss occurs in converting the Ihp. of the air engine cylinder to shaft hp., this being due to mechanical friction of the air engine. The thermodynamic process would start with the heat in the steam entering the steam cylinder of the engine which drives the compressor. As the efficiency of the steam engine under average conditions is around 0.10, the efficiency from steam to Ihp. of air engine is about $0.10 \times 0.391 = 0.0391$: less than 4 per cent.

The preheater "boosts" the air engine output to a moderate extent, and the efficiency of its part of the process has been shown, in one illustrative case, to be 32.6 per cent. Thus its effect is to improve the efficiency of the whole system, although it is not installed primarily for that purpose. Compressed air is unique among transmission systems, in that supplementary energy can be supplied at the receiving end with highly beneficial influence on efficiency.

56. **Tests.** Observed air rates in well-designed engines are from 477 to 1000 cu. ft. of free air per hour per brake hp. Small engines without expansion use around 2000 cu. ft.

The probable fall of pressure in the pipe line may be estimated from Unwin's formula:

$$p_2 = p_1 \sqrt{1 - \frac{fu^2G}{430Td}}, \quad \text{in which} \tag{35}$$

p_2 and p_1 are pressures at end and beginning of line, lb. per sq. in., absolute,

f = 0.00435 for 6 in. pipe, 0.004 for 8 in. pipe and 0.003 for pipe 12 in. or larger,

u = velocity of air in pipe, ft. per sec.

$$= \frac{\text{volume}}{\text{area}} = \left(\frac{F}{60} \times \frac{14.7}{p_1}\right) \div \frac{\pi}{4}d^2,$$

G = length of pipe line, ft.,

T = absolute temperature of air,

d = internal diameter of pipe, ft.

Prob. 104. A 12-in. pipe line 5000 ft. long carries 10 000 cu. ft. of free air per min., the initial pressure being 147 lb. per sq. in. Find the volume and velocity per sec. of air flowing through the pipe. (Ans., vol. = 16.67 cu. ft., u = 21.25 ft.)

Prob. 105. If the temperature of this air is 40° F., find the pressure at the end of the pipe line. (Ans., 145 lb. per sq. in.)

Prob. 106. If T_g, Fig. 17, is 840°, find t_h. (Ans., 201° F.)

Air Refrigeration

57. **Joule Cycle Reversed.** The fall of temperature which occurs during expansion in the cylinder of an air engine may be (and has been) employed for refrigeration. In Fig. 14, suppose the Joule cycle *abcd* to be worked counter clockwise. Heat is ABSORBED from *d* to *c*, the fluid is compressed (the temperature rising) from *c* to *b*, heat is EMITTED from *b* to *a* and the fluid is expanded (the temperature falling) from *a* to *d*. (Compare Art. 35.) The operation *dc* is a REFRIGERATION of surrounding bodies.

During this operation, the fluid is circulating through pipes placed in a brine tank or room to be cooled.

Refrigeration being the aim in view, the efficiency must be regarded as

$$e = \frac{H_{dc}}{\Sigma W} = \frac{H_{dc}}{H_{ba} - H_{dc}} = \frac{k(T_c - T_d)}{k(T_b - T_a) - k(T_c - T_d)}$$

$$= \frac{T_c - T_d}{T_b - T_a - T_c + T_d}.$$

Since the curves bc and ad are isodiabatics,

$$\frac{T_b}{T_a} = \frac{T_c}{T_d}, \quad \frac{T_b}{T_a} - 1 = \frac{T_c}{T_d} - 1, \quad \frac{T_b - T_a}{T_c - T_d} = \frac{T_a}{T_d},$$

$$\frac{T_b - T_a - T_c + T_d}{T_c - T_d} = \frac{T_b - T_a}{T_c - T_d} - 1 = \frac{T_a}{T_d} - 1 = \frac{T_a - T_d}{T_d}$$

and

$$e = \frac{T_d}{T_a - T_d} = \frac{T_c}{T_b - T_c} = 1 \div \left\{ \left(\frac{p_b}{p_c} \right)^{(\nu-1)/\nu} - 1 \right\}, \quad (36)$$

which may of course exceed unity, as in Art. 35.

The comparable efficiency of the Carnot cycle would be based on the range of temperatures DURING THE REFRIGERATING PROCESS ALONE, i.e., along dc, and would be $T_d \div (T_c - T_d)$. This exceeds the value by Equation (36).

Prob. 107. In a Joule cycle for refrigeration, the compression ratio ($C = p_a/p_d$, Fig. 14) is 4. Find the efficiency if $y = 1.406$. (Ans., 2.04.)

Prob. 108. If in Prob. 107 and Fig. 14, $T_d = 420$, $T_c = 571$, find T_a, T_b and the efficiency of refrigeration in the CARNOT cycle. (Ans., $T_a = 628°$, $T_b = 855°$, $e_C = 2.78$.)

58. **Ideal Air Machine.** In Fig. 14, the diagram $fcbe$ may be assumed to represent the action of a compressor, $eadf$ that of an engine. The area $abcd$ represents the difference between the power absorbed by the compressor and that generated by the engine. In an ice machine, power equivalent to this area, plus the power necessary to overcome friction of mechanism, must

be provided from a separate source, such as a steam engine. Let subscripts $_C$, $_E$, and $_S$ refer to compressor, air engine and steam engine, respectively. Then for 1 lb. of air,

$$abcd = 778(H_{ab} - H_{cd}) = 778\{k(T_b - T_a) - k(T_c - T_d)\},$$

$$\frac{W_C - W_E}{W_E} = \frac{abcd}{W_E} = \frac{W_C}{W_E} - 1,$$

$$W_E = W_{ea} + W_{ad} - W_{df} = p_e(V_a - V_e) + \frac{p_aV_a - p_dV_d}{y - 1} - p_d(V_d - V_f)$$

$$= (p_aV_a - p_dV_d)\frac{y}{y - 1} = \frac{Ry}{y - 1}(T_a - T_d),$$

$$\frac{W_C}{W_E} = 1 + \frac{778k(T_b - T_a - T_c + T_d)}{Ry(T_a - T_d)} \cdot (y - 1),$$

or since

$$R = \frac{k}{y}(y - 1) \times 778,$$

$$\frac{W_C}{W_E} = 1 + \frac{T_b - T_c}{T_a - T_d} - 1 = \frac{T_c}{T_d} = \frac{V_c}{V_d} = \frac{D_C}{D_E}.$$

Also, since $W_S = W_C - W_E$,

$$W_S = W_E\left(\frac{V_c}{V_d} - 1\right) = W_E\left(\frac{D_C}{D_E} - 1\right),$$

where D = DISPLACEMENT.

Prob. 109. Under the conditions of Prob. 108, what is the ratio of air engine power to compressor power? Of steam engine power to compressor power? (Ans., 0.735, 0.265.)

59. **Ideal Power and Units.** The heat absorbed along dc, in Fig. 14, for the temperatures assigned in Prob. 108, is

$H = k(T_c - T_d) = 0.2375 \times 151 = 35.9$ B.t.u. per lb. of air.

The unit of refrigerating capacity is "ICE-MELTING EFFECT." If water exists at 32° F., then under purely ideal conditions, the abstraction of 144 B.t.u. from 1 lb. of this water will convert it to ice. Conversely, the addition of 144 B.t.u. to 1 lb. of ice

will liquefy it. The quantity 144 is the LATENT HEAT OF FUSION of ice. The "ice-melting effect" of a refrigerating machine, in lb., is the NUMBER OF B.T.U. IT ABSORBS along *dc*, Fig. 14, DIVIDED BY 144. Suppose the ice-melting effect required is 113 lb. PER HR. Then the refrigeration to be done per min. is $(113 \times 144) \div 60 = 271$ B.t.u. If this is to be done when, as above, 35.9 B.t.u. are removed per lb. of air, the weight of air to be circulated per min. is $271 \div 35.9 = 7.54$ lb.

Efficiency is refrigeration divided by the heat equivalent of power expended. Its value here (Prob. 107) is 2.04. Hence since 1 hp. = 42.42 B.t.u. per min., the steam cylinder hp. necessary is $(271 \div 2.04) \div 42.42 = 3.16$ and (from Prob. 109) the air cylinder will develop $(0.735/0.265) \times 3.16 = 8.8$ hp. while the compressor cylinder will consume $8.8 + 3.16 = 11.96$ hp. In symbols, let Q = B.t.u. of refrigeration done per minute. Then steam cylinder hp. $= Q \div 42.42e$, ice-melting effect $= Q \div 144$ lb. per min., lb. of air circulated per min. $= Q \div k(T_c - T_d) = w_a$.

Heat must be removed from the air along *ba*, Fig. 14, in an external "cooler" or "condenser." The heat removed per min. is $q = w_a k(T_b - T_a)$. (It should be noted that steam cylinder hp. $= (q - Q) \div 42.42$, if the curves *bc* and *ad* are adiabatic.)

The TONNAGE of a machine is the ice-melting effect per 24 hr., expressed in tons of 2000 lb. If T = tonnage,

$$T = \frac{60Q \times 24}{2000 \times 144} = \frac{Q}{200}.$$

A unit of economical performance is the ICE-MELTING EFFECT PER HR. PER IHP. OF STEAM CYLINDER. Call this I_0. Then

$$I_0 = \frac{60Q}{144} \div \frac{Q}{42.42e} = 17.6e. \tag{37}$$

Prob. 110. From Prob. 108, find q if 7.54 lb. of air are circulated per min. (Ans., 407 B.t.u.)

Prob. 111. What is the Ihp. of the steam cylinder computed from $q = 407$ and $Q = 271$? (Ans., 3.2.)

Prob. 112. For $Q = 271$, what is the tonnage? (Ans., 1.355.)

Prob. 113. Find I_0 for the conditions of Art. 59. (Ans., 35.9.)

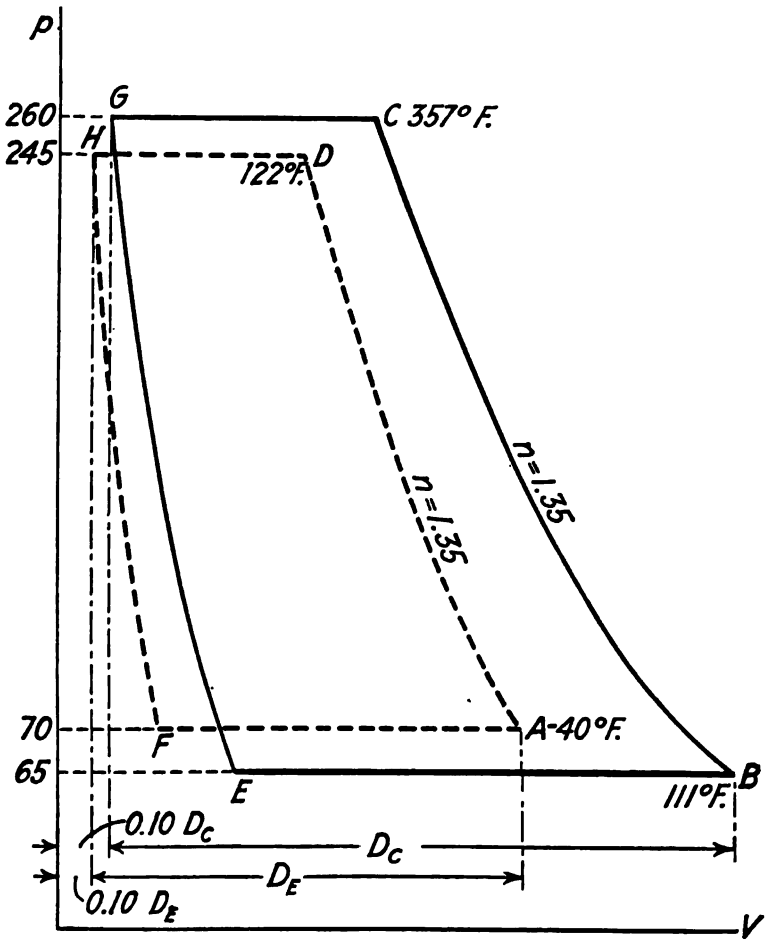

FIG. 18. Dense Air Ice Machine.

60. **Dense Air Ice Machine.** Fig. 18 refers to the action of an actual ice machine, which consists of compressor and air engine (expander) cylinders side by side, with a steam cylinder. The action of the steam cylinder is not shown by the diagram. The

compressor cycle is *EBCG*, the pressure limits noted being approximately those of practice and the compression ratio 4, as used for the Joule cycle, Prob. 107. The engine (expander) cycle is *HDAF*, the pressure differences between *GC* and *HD* and between *AF* and *EB* being somewhat exaggerated. The curves are NOT adiabatic. We will take $n = 1.35$. The compressor should be jacketed like any compressor. This reduces the work of the condenser, which must cool the air from *C* to *D*. The air engine cycle is given "complete" expansion, in order that the fall of temperature may be a maximum.

The air leaving the engine goes to the brine tank or other place to be cooled, then returning to the compressor. The necessary value of t_A is determined by the service. We will assume $t_A = -40°$ F., whence $T_A = 420$. T_B will be higher. If ice is to be made, it cannot be much above 0° F. (460). It will exceed the cold room temperature somewhat, because the air will receive heat from the atmosphere on its way from the cold room to the compressor.

In a specific machine, the compressor and expander cylinders were $5\frac{3}{4}$ by 10 and $4\frac{3}{4}$ by 10 inch and connected to the same shaft. The ratio of displacements is then $D_C/D_E = (5.75/4.75)^2 = 1.46$.

$$\frac{p_A V_A}{T_A} = \frac{p_B V_B}{T_B}, \quad \frac{T_B}{T_A} = \frac{p_B V_B}{p_A V_A} = \frac{65}{70} \cdot \frac{D_C}{D_E} = 1.36,$$

both clearances being 10 per cent. Then $T_B = 420 \times 1.36 = 571$ or 111° F. This is an unusually high value, but we will adopt it, as in Prob. 108.

Prob. 114. Find refrigeration per lb. of air in Fig. 18. Note: consider the pressure as constant during heating from *A* to *B*. (Ans., 35.9 B.t.u.)

Prob. 115. Compute T_D, T_C. (Ans., 582°, 817°.)

The value of T_D (122° F.) is easily realized at the condenser: which should in fact be designed to reduce this temperature to 80°–100° F. The heat to be carried away at the condenser is $0.2375(T_C - T_D) = 55.9$ B.t.u. per lb. of air.

The value of D_C is $(\pi/4) \times (5.75)^2 \times (10/1728) = 0.15$ cu. ft. per stroke. The volume of air drawn into the compressor (and therefore circulated through the system) per stroke is $V_B - V_E = 0.82D_C = 0.123$ cu. ft. (see Prob. 116). The weight of air per stroke is $(P_E \times 0.123) \div (53.36T_B) = (65 \times 144 \times 0.123) \div (53.36 \times 571) = 0.0377$ lb. Assume $N = 100$, i.e., 200 single strokes per minute, the machine to be double-acting. Then the weight of air circulated per min. is $w_a = 200 \times 0.0377 = 7.54$ lb. (Compare Prob. 110.) We then have

$$Q = 35.9 \times 7.54 = 271 \text{ (see Art. 59 and Prob. 114),}$$

$$q = 55.9 \times 7.54 = 421.$$

In this actual engine, the steam cylinder hp. is not determined by $q - Q$, since heat transfers occur along all the curved paths of the cycles. The ihp. of each cylinder (compressor and expander) is first worked out in detail. Applying Equation (29) to the compressor,

$$p_{mC} = \frac{1.35 \times 65}{0.35}\{1.1(4^{0.259} - 1) + 0.10(4^{0.741} - 4)\} = 88.6$$

and to the engine (expander),

$$p_{mE} = \frac{1.35 \times 70}{0.35}\{1.1(3.5^{0.259} - 1) + 0.10(3.5^{0.741} - 3.5)\} = 88.$$

The corresponding horse powers are

$$\text{Ihp.}_C = \frac{2 \times 88.6 \times 10 \times 0.7854 \times 5.75 \times 5.75 \times 100}{12 \times 33\,000} = 11.6.$$

$$\text{Ihp.}_E = \text{Ihp.}_C \times \left(\frac{4.75}{5.75}\right)^2 \times \frac{88}{88.6} = 7.8.$$

(Powers at equal piston speeds are proportional to the products of area and p_m.) Then the net power to be derived from the steam cylinder is $11.6 - 7.8 = 3.8$. To this we may add 50 per cent. to cover friction losses, so that $\text{Ihp.}_S = 5.7$. These results may be compared with those for the ideal cycle computed in Art. 59. It should be noted that compression in the air engine is here "complete," i.e., the curve FH, Fig. 18, continues until the air-supply pressure is attained.

The efficiency is in this case

$$\frac{Q}{42.42 \times 5.7} = \frac{271}{241} = 1.13.$$

For the ideal cycle, it was 2.04. The tonnage is that of the ideal cycle, 1.355. The ice-melting effect per Ihp.-hr. is $I_0 = 17.6 \times 1.13 = 19.8$ lb. The value realized for this depends mainly on the temperature range. In practice, the useful value of Q is much reduced by radiation, and values of e around 0.5 or 0.6 or of I_0 around 10 are common.

If the machine were to make ice from water at 100°, it would have to cool the water well below the freezing point, first,—say to 0°. This would require the extraction of approximately 100 B.t.u. per lb. Adding the latent heat of fusion, the refrigeration necessary per lb. of ice made is 244 B.t.u. The machine considered could then make a maximum of 271/244 = 1.11 lb. of ice per min. During the preliminary cooling, it could lower the temperature of 271 lb. of water 1° per min.

Prob. 116. In Fig. 18, find V_E and $V_B - V_E$.
(Ans., $0.28D_C$ and $0.82D_C$.)

Prob. 117. In Fig. 18, find V_C. (Ans., $0.393D_C$.)

Prob. 118. In Art. 60, check the values of p_{mC} and p_{mE}. Calculate Ihp.$_E$ by the formula $2p_mLAN \div 33\,000$.

Regenerative Hot-Air Engine Cycles

61. **Remarks.** As has been stated, the air engine of a compressed air plant is not a thermal engine but only a part of a work-conversion system. Another type of engine, which uses high-temperature air, discharging it at a lower temperature, is now to be considered. This is really a thermodynamic machine. Such engines are called HOT-AIR ENGINES. They are NOT internal combustion engines, for fuel does not enter the cylinder. They differ from most engines (not from dense air ice machines) in that a fixed mass of fluid is used over and over again. The use of the REGENERATOR, common with hot air engines, makes the ideal efficiency of their cycle EQUAL TO THAT OF THE CARNOT

CYCLE. In this respect they are unique. The regenerator is a closed vessel filled with wire gauze, glass or any substance of large heat absorbing capacity.

62. **Stirling Cycle.** Fig. 19 shows the principle of operation of the Stirling engine. The piston being at the extreme left,

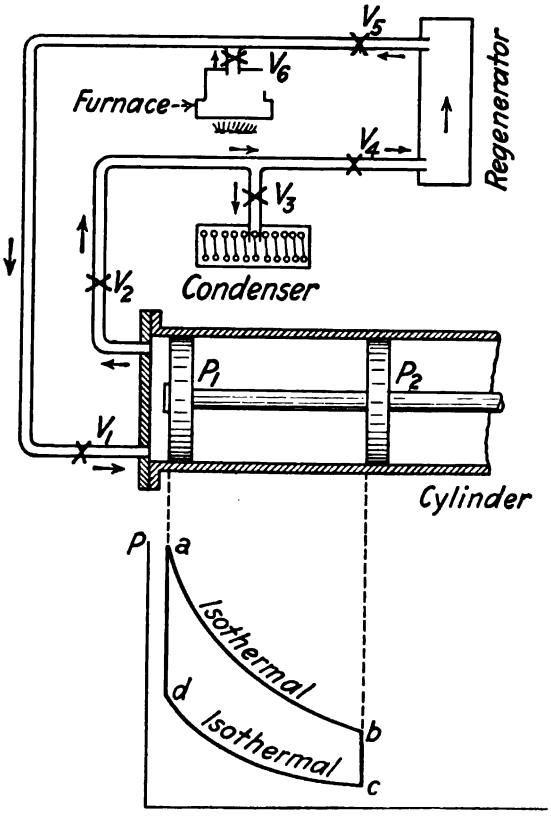

FIG. 19. Hot Air Engine with Regenerator—Stirling Cycle.

position P_1, the state on the lower diagram is d. Valves V_2 and V_6 are closed, valves V_1 and V_5 are open. Hot air from the regenerator flows into the cylinder, the temperature rising to T_a and the pressure increasing as long as the piston does not move. Valve V_5 is now closed and V_6 is opened. Heat from the furnace

causes the air in the cylinder to expand (path *ab*) and the piston moves to the extreme right (position P_2). The supply of heat is so controlled with relation to the piston speed that the path *ab* is ISOTHERMAL. The valve V_1 is now closed, V_2 and V_4 are opened and V_3 closed. Air flows from the cylinder to the regenerator. The pressure in the cylinder falls (path *bc*), the piston remaining stationary. As the air flows into and through the regenerator, its temperature falls: but the inlet temperature at the regenerator finally approximates T_b, while its outlet temperature approximates T_c. Valve V_4 is now closed and V_3 opened. Then the piston makes a full stroke to the left, forcing air into the condenser. The water supply in the condenser coils is so controlled as to withdraw heat without lowering the temperature. The path *cd* is consequently ISOTHERMAL.

Under purely ideal conditions, all heat discharged to the regenerator along *bc* is regained along *da*, so that this quantity of heat may be disregarded as a debit against the engine. The only heat chargeable is H_{ab}, and the efficiency is

$$e = (H_{ab} - H_{cd}) \div H_{ab} = \frac{T_a - T_d}{T_a}$$

(Art. 34), which is the EFFICIENCY OF THE CARNOT CYCLE.

The operation of the ERICSSON hot-air engine was similar, and its efficiency is the same, but the regenerative processes were at CONSTANT PRESSURE instead of at constant volume.

For $T_{ab} = 1000$, $T_{cd} = 500$, $e = 0.50$. WITHOUT THE REGENERATOR, the efficiency would have been

$$e = \frac{W_{ab} - W_{dc}}{H_{da} + H_{ab}} = \frac{RT_a \log_e \frac{V_b}{V_a} - RT_d \log_e \frac{V_c}{V_d}}{778l(T_a - T_d) + RT_a \log_e \frac{V_b}{V_a}}$$

$$= \frac{(T_a - T_d) \log_e J}{\frac{778l}{R}(T_a - T_d) + T_a \log_e J},$$

in which $J = V_b/V_a$ and (since *ab* and *dc* are isodiabatics) also

$= V_c/V_d$. For the temperatures above assumed, this becomes

$$e = \frac{1150 \log J}{1232 + 2300 \log J} = \frac{\log J}{1.07 + 2 \log J}.$$

Then e increases as J increases.

The mean effective pressure, with or without regenerator, is

$$p_m = \frac{W_{ab} - W_{dc}}{144(V_c - V_d)} = \frac{R \log_e J(T_a - T_d)}{144V_d(J - 1)}.$$

For the assumed conditions, this is

$$p_m = \frac{53.36 \times 2.3 \times 500 \log J}{144V_d(J - 1)} = \frac{426 \log J}{V_d(J - 1)}.$$

This decreases (for a fixed value of V_d) as J increases. Hence low mean effective pressures accompany high efficiencies.

Hot-air engines with regenerators have given up to 25 per cent. efficiency even in small sizes, and have used as little as 1.7 lb. of coal per Ihp. per hr., but the mean effective pressures are low and the heat-transmitting surface at the furnace burns out rapidly.

Prob. 119. In Art. 62, find efficiencies without regenerator for $J = 4, 5, 10, 20, \infty$. (Ans., 0.27, 0.28, 0.33, 0.36, 0.50.)

Prob. 120. Find $V_d p_m$ in the above.

(Ans., 86, 75, 47, 28, indeterminate.)

CHAPTER IV

STEAM AND OTHER VAPORS

Vapor Properties and Tables

63. **Heating of Liquid.** Consider the fundamental equation, $H = T + I + W$, Art. 2, Equation (3). For liquids and their vapors, I (the disgregation work) is NOT equal to zero. The equation is usually written for such fluids, $H = E + W$, where $E = T + I$ and comprises all effects of heat other than external work, and may be called the INTERNAL WORK, or gain of INTERNAL ENERGY.

When 1 lb. of liquid at 32° F. is heated, a rise of temperature and (in general) an increase of volume occur. W is therefore positive. Its value is very small. There are few cases, in the instance of water, for example, where it exceeds 1 B.t.u. Hence in the heating of water E is nearly equal to the whole amount of heat supplied (see Art. 77).

The heating of the liquid may be continued until it begins to boil. At atmospheric pressure, each liquid has its own definite boiling point. At a lower pressure, it will boil at a lower temperature. At a higher pressure, it will boil at a higher temperature. Fig. 20 shows the temperatures at which boiling occurs, for various liquids, at various pressures. Water appears here (with one exception) as having the highest boiling temperatures, but there are many liquids LESS volatile than water. All curves show that as the temperature increases, the pressure increases more and more rapidly. Therefore if high temperatures are desired in a steam engine (so as to attain high Carnot efficiency) high pressures almost necessarily result. There is an exception to this, however: see Art. 83.

The HEAT OF THE LIQUID IS THE QUANTITY OF HEAT NECESSARY TO RAISE THE TEMPERATURE OF 1 LB. OF LIQUID FROM

32° F. TO THE BOILING TEMPERATURE. Its symbol is h. The boiling temperature depends upon the pressure at which boiling occurs. There is a definite temperature of boiling for every

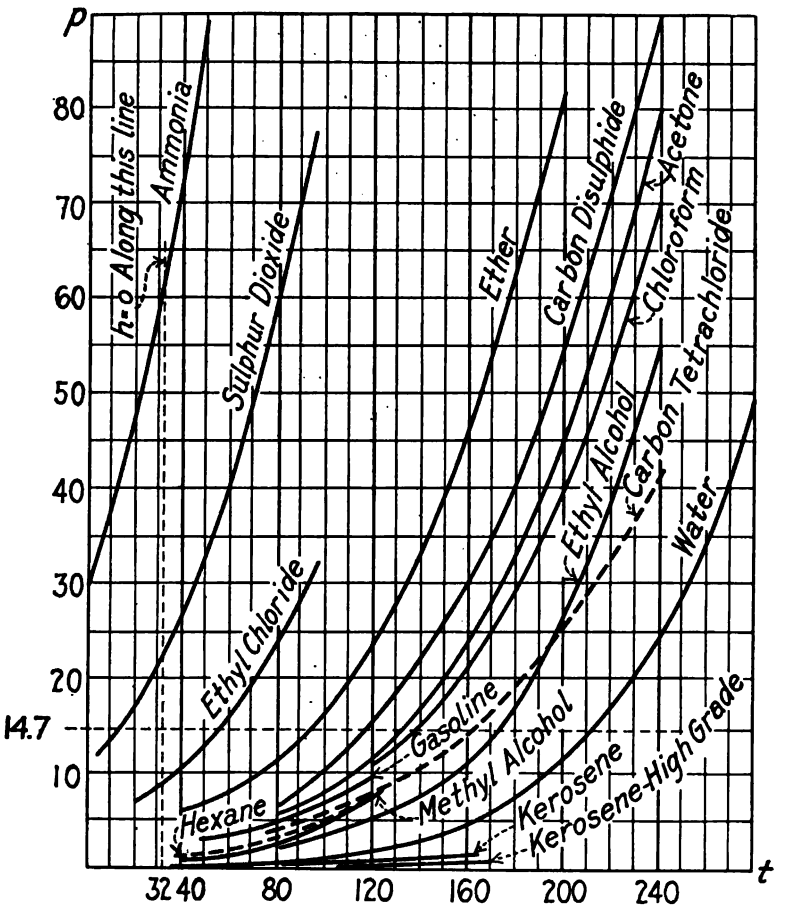

FIG. 20. Boiling Temperatures for Various Pressures.

pressure (Fig. 20) of a given liquid.* The temperature 32° F. is arbitrarily taken as a starting point. The value of h for all

* As tabulated on pages 82, 83, values of h consider the pressure at the start to be that pressure at which the boiling point is 32°, viz., 0.0886 lb. per sq. in. The pressure then increases as the water is heated, until the boiling s reached.

liquids is at this temperature, consequently, zero. For every different pressure at which boiling occurs, there will be a different value of h.

The boiling point of ammonia at atmospheric pressure is $-27°$ F. This is BELOW 32° F., hence the value of h at this temperature is NEGATIVE. If t = temperature at which boiling occurs, s = mean specific heat of liquid from 32° to $t°$, $h = s(t - 32)$. The value of s is generally somewhat variable.

Prob. 121. Compute h for water when $t = 212°$.

(Ans., 180 B.t.u.)

Prob. 122. When water boils under 100 lb. absolute pressure, its temperature is 327.8° and $h = 298.3$. Find s from 32° to 327.8°. (Ans., 1.008.)

Prob. 123. The mean specific heat of liquid ammonia from 5° to 32° is 1.052. Find h at 5°. (Ans., -28.4 B.t.u.)

64. **Vaporization.** If more heat is added after the liquid reaches its boiling temperature, VAPOR is formed. No vapor can be formed until the liquid has reached boiling temperature. During the formation of the vapor, it and the liquid are at the same temperature. This temperature is that at which the liquid boils at the existing pressure. The temperature of the vapor cannot be increased above this as long as any liquid is present. The removal of heat from a vapor (if kept at constant pressure) does not lower its temperature, but liquefies it, or part of it.

A pound of liquid completely evaporated makes a pound of vapor. The vapor has a definite SPECIFIC VOLUME, v. Its temperature we have shown to be t. The whole operation of converting the pound of already boiling liquid into vapor consumes or requires the heat L, defined as the LATENT HEAT OF VAPORIZATION, all of which is supplied at CONSTANT TEMPERATURE, t, of the fluid.

The kind of vapor thus formed is called SATURATED VAPOR or DRY VAPOR. At a given pressure, saturated vapor has the lowest possible temperature and specific volume of any vapor. WET VAPOR may be regarded as incompletely evaporated liquid.

That is, for example, from 1 lb. of liquid at the boiling point, ½ lb. of saturated vapor may have been formed, so that the resulting fluid is a mixture of equal parts of vapor and boiling liquid. The temperature and specific volume of the vapor in this mixture will be t and v. SUPERHEATED VAPOR is formed by adding still more heat to saturated vapor. This can only be done after all of the liquid is evaporated. The temperature, t', of superheated vapor is always higher than t: its volume, v', is greater than v. It behaves like an imperfect gas.

Prob. 124. A saturated vapor exists at the pressure $p = 100$ and the temperature $t = 327.8$. The pressure is increased while the temperature remains constant. What happens? (Ans., the vapor must liquefy.) If the pressure is LOWERED while the temperature remains constant, what happens? (Ans., the vapor becomes superheated.)

65. **Variation of L.** In general, L decreases at high pressures. This does not mean that it is easier to generate vapor at high pressure, because as the pressure increases, the boiling temperature increases and hence h increases. The increase in h is more than the decrease in L, in the case of steam. The total heat necessary to generate dry saturated vapor from 1 lb. of liquid at 32° is called the TOTAL HEAT, H, and is equal to $h + L$. This quantity increases, for steam, as the pressure increases. It always requires more heat to transform water (at a given initial temperature) to dry steam, when the pressure is high. The amount of heat required to vaporize liquid ALREADY AT THE BOILING POINT decreases with increase of pressure. This quantity we call L.

At moderate pressures and for most vapors, L is greater than h. Thus for water boiling at 212° F., $L = 970.4$ while $h = 180$. As the pressure increases, h increases and L decreases. Finally a pressure is reached at which $L = 0$. The temperature at which boiling then occurs is called the CRITICAL TEMPERATURE. This, for steam, is 689° F. At the critical temperature, the liquid (brought to the boiling point) passes to the condition of saturated

vapor without any further addition of heat. The liquid and vapor states thus merge into each other. If a liquid is heated above its critical temperature it becomes a gas. It cannot again be liquefied by mere increase of pressure until its temperature has first been reduced below the critical point. This is of importance in connection with carbon dioxide, which has a critical temperature around 88° F. This substance is used as a fluid in refrigerating plants on some vessels. The refrigerating fluid must be condensed in one part of the operation. If CO_2 is used, its temperature must then be brought below 88°. Conditions may arise in tropical waters where this would be impossible. The CO_2 refrigerating machine cannot then be used.

In the equation, heat $= E + W$ (Art. 63), as applied to vaporization ALONE, how much of the heat of vaporization, L, is an E effect, and how much a W effect? We are considering vaporization alone: the liquid is already at the boiling point: $L = E + W$. Of the heat added during vaporization, W units are used in performing external work, and E units in performing internal work, or increasing the internal energy. Take the case of steam formed from water at 400°, and therefore (by the steam table) at 247.1 lb. pressure. The specific volume of the water at 400° is 0.0187. During vaporization, this is increased to $v = 1.872$. The external work part of L is therefore W_L = pressure × increase of volume =

$$\frac{144 \times 247.1(1.872 - 0.0187)}{778} = 84.2 \text{ B.t.u.}$$

The E effect must be $L - 84.2$. The value of L given by the steam table is 827.2. Hence the E effect is 743: much larger than the W effect. The symbol r is used for the E effect included in L. Since h is nearly all E effect (Art. 63), $H = h + L$ consists approximately of an E effect equal* to $h + r$ and a W effect nearly equal to $144p(v - v_w) \div 778$, where v_w = specific volume of liquid at the boiling point.

* The EXACT value is given in Art. 77.

Prob. 125. Using the steam table, pages 228, 229, take values of p, v, L, as given at 327.8° F., and check the value given for r. Note $v_w = 0.01774$ from Fig. 22*a*.

66. **The Steam Table.** The steam table, given on pages 228–229, is a typical vapor-property table. Similar tables exist for other vapors like ammonia, SO_2, CO_2: but the properties of many vapors are only imperfectly established. The basis or "argument" of the table is either the PRESSURE or the TEMPERATURE. For a given value of the pressure, p, t is the Fahrenheit temperature at which boiling occurs. The specific volume of the vapor is v, which is less for high than for low pressures. The heat quantities are for 1 lb. of fluid, originally liquid at 32° and at 0.0886 lb. pressure. The meaning of h, L, H and r has been explained. All tables have an elaborate experimental basis, and for saturated steam, the differences between the various existing tables are of small importance. The value of H here used is based on the Davis formula,

$$H = 1150.3 + 0.3745(t - 212) - 0.00055(t - 212)^2. \quad (38)$$

Prob. 126. From the Davis formula, find H when $t = 212°$. (Ans., 1150.3.)

Prob. 127. Compute H when $t = 312°$. (Ans., 1182.25.)

Prob. 128. From the table, what is the mean specific heat of water between 32° and 312°? (Ans., 1.007.)

Prob. 129. What is the approximate value of E in the formation of 1 lb. of steam at 80 lb. pressure from water at 32° F.? (Ans., 1101.8.) What is the value of W during vaporization? (Ans., 80.5 B.t.u.)

Entropy

67. **A New Diagram.** Art. 33 gave a graphical method of representing HEAT ABSORBED, of which we have not made much use. The subtended area under a path on the PV diagram, representing EXTERNAL WORK DONE, has on the other hand been used repeatedly. The magnitude of this area for specific processes is easily computed, because three of its boundaries are

straight lines. We are now to develop a similarly simple area for the representation of QUANTITIES OF HEAT absorbed or emitted. The new diagram may be conceived as having been produced by a mechanism " interlocked " with the indicator, so that while

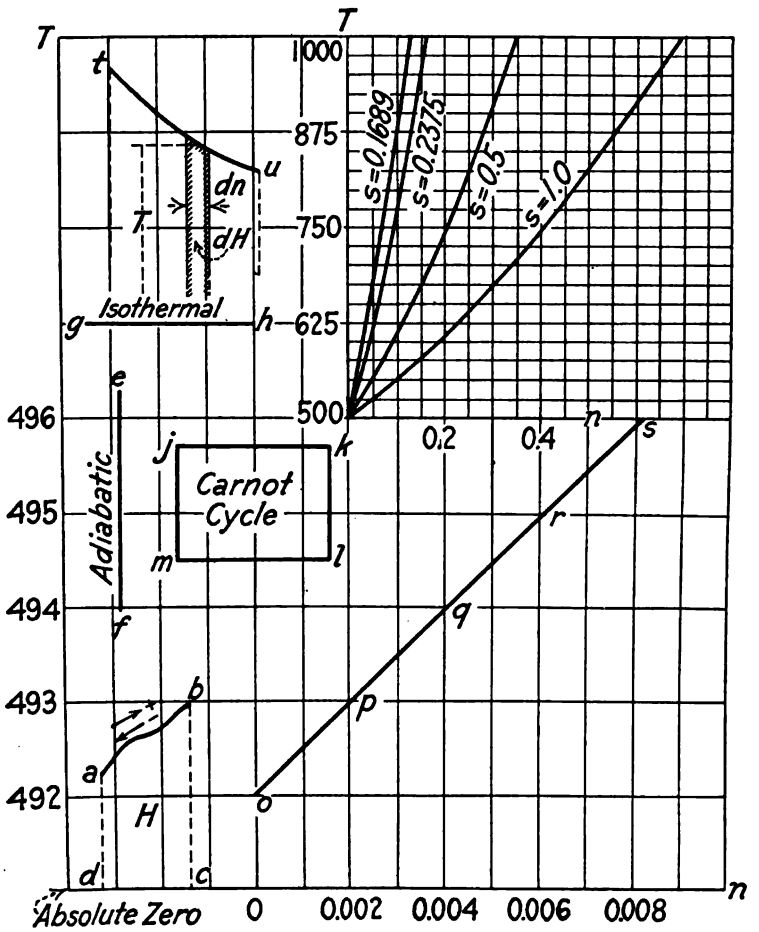

FIG. 21. Entropy Diagrams.

the pencil of the latter is describing lines the areas under which represent external WORK, the imaginary pencil of the new diagram will simultaneously sketch corresponding lines, areas under which will represent HEAT.

The ordinate of the diagram is to be absolute TEMPERATURE. In Fig. 21, then, T is the ordinate and *abcd* represents the heat absorbed or emitted along *ab*, or briefly $abcd = H_{ab}$. If *ab* is described from left to right, H_{ab} is $+$: if from right to left, H_{ab} is $-$. Bearing in mind that SUBTENDED AREAS REPRESENT HEAT and ORDINATES ARE ABSOLUTE TEMPERATURES, answer the following:

Prob. 130. What is the shape of an isothermal on this diagram?

Prob. 131. What is the only possible shape of an adiabatic? Why?

Prob. 132. What is the shape of a Carnot cycle? Sketch and letter the areas representing H_{jk} and H_{lm}. Which is plus and which minus if the cycle is clockwise? What area represents the net external work done in the Carnot cycle? ($\Sigma W = \Sigma H$ = heat received $-$ heat rejected.) Efficiency is ΣH divided by all of the $+$ heat. In this case, it is the quotient of what two areas? Of what two lengths (in rectangles of equal width, areas are proportional to lengths). State the efficiency in terms of T_{jk} and T_{ml} and prove the statement.

68. **The Abscissa of the Diagram.** Consider the heating of 1 lb. of water from 32° F. or 492° absolute. Heat it 1°. The heat absorbed is 1.01 B.t.u. (the specific heat of water is not exactly 1.0). Let *o*, Fig. 21, represent the condition of the water before heating. A path representing the heating process must be subtended by an area of 1.01 B.t.u. If we assume the path to be a straight line, the subtended area is a trapezoid of area 1.01 and parallel vertical sides of lengths 492 and 493 (absolute temperatures). Its horizontal width is therefore AREA $\div$ MEAN HEIGHT $= 1.01 \div 492.5 = 0.00205$. The process is represented by the line *op*: the coördinates of *o* being 492, 0 (the latter fixed as an arbitrary starting point) and those of *p* being 493, 0.00205.

Heat this water another degree. The heat supplied is this time 1.0 B.t.u., the mean temperature 493.5, the horizontal

width of the trapezoid 0.00203, and the coördinates of q are 494 and 0.00205 + 0.00203 = 0.00408.

Prob. 133. In heating from 34° to 35° F., and also from 35° to 36° F., 1.01 B.t.u. are supplied. Locate points r and s, Fig. 21. (Ans., abscissas are 0.00613 and 0.00817.)

The whole operation from 32° to 36° is represented by os, which is nearly a straight line. This method is approximate, involving the assumption that each of the segments op, pq, qr, rs, is a straight line. The error due to this is exceedingly small.

Consider the process tu. The area under this curve represents H_{tu}. Consider a very small portion of the curve, along which the temperature may be assumed to be constant. Call this temperature T. The narrow area under the small portion of the curve is dH and the width of this area is dH/T. The horizontal component or abscissa of the whole path tu is then

$$n_{tu} = n_u - n_t = \int \frac{dH}{T}, \text{ between proper limits.}$$

The dimension n is called the ENTROPY. THE ONLY DEFINITION OF ENTROPY IS $\int(dH/T)$. It is the abscissa of our diagram, which is accordingly marked n. There is no necessary origin for entropy. The line os puts it at 32° F. or 492° absolute, but this need not always be the case. Only CHANGES of entropy are important. The symbol n_{tu} or the quantity $n_u - n_t$ both mean the CHANGE OF ENTROPY or the HORIZONTAL COMPONENT OF THE PATH from t to u. We must now establish specific values for $\int(dH/T)$ for various processes.

69. **Change of Entropy at Constant or Definite Specific Heat.** For the path tu, $n_u - n_t = \int(dH/T)$. Suppose this path to be one along which the specific heat is constant = s. Then $H = s(T_2 - T_1)$ for 1 lb. weight heated from T_1 to T_2 or $dH = s\,dT$. Then

$$n_u - n_t = \int_t^u \frac{s\,dT}{T} = s(\log_e T_u - \log_e T_t) = s \log_e \frac{T_u}{T_t}. \quad (39)$$

Note that T_u and T_t are ABSOLUTE temperatures. Suppose the specific heat to be variable: for example, $s_1 = a + bT$. Then

$$n_u - n_t = \int_t^u \frac{a dT}{T} + \int_t^u b dT = a \log_e \frac{T_u}{T_t} + b(T_u - T_t).$$

The case of water comes in neither of these classes. Its specific heat is too irregularly variable to be expressed as a linear function of the temperature. An approximation can be made, for water, by taking s as the mean value over the temperature range specified, in Equation (39).

Thus for water heated from 32° to 36° F. the heat expenditure is 4.03 B.t.u. Take s, then, at $4.03 \div 4 = 1.0075$. Then by Equation (39),

$$n_{36} - n_{32} = 1.0075 \times 2.3026^* \log \frac{496}{492} = 0.0082.$$

This result is comparable with that reached in Prob. 133 and used in plotting the point s, Fig. 21. It should be noted that with a positive specific heat the path must slope upward to the right and downward to the left: that low values of the specific heat lead to STEEP paths: and that any path of constant specific heat becomes steeper as it ascends.

Prob. 134. Given $T_u = 2T_t$. Find n_{tu} when $s = 0.1689$, 0.2375, 0.5, 1.0. (Ans., 0.117, 0.164, 0.347, 0.693.)

Prob. 135. What is the gain of entropy in heating 1 lb. of water from 32° F. to 312° F.? Note: take $s = h \div (t - 32)$.

(Ans., 0.4535.)

70. **Entropy at Various Specific Heats.** For an adiabatic, $s = 0$ and the path is vertical. The table in Art. 29 mentioned the adiabatic as a path of constant entropy. For an isothermal, $s = \infty$ and the path is horizontal. For all constant values of s between 0 and $+\infty$, the curve is a LOGARITHMIC CURVE. Several paths are plotted in the small diagram in the upper right-hand corner of Fig. 21. Equation (39) may be used to determine either end-points or intermediate points of these paths.

Prob. 136. If $s = 1$, $T_t = 500$, $T_u = 750$, find n_{tu}.

(Ans., 0.406.)

* The use of the modulus permits employing a table of common logarithms instead of Napierian logarithms.

71. **Entropies of Vapor.** The ENTROPY OF THE LIQUID, n_w, is the CHANGE OF ENTROPY DUE TO THE HEATING OF 1 LB. OF LIQUID FROM 32° F. TO THE BOILING POINT. It was computed approximately for water with $t = 312°$ F. in Prob. 135. Similar computations taken for successive short ranges of temperature give all of the values of n_w which appear in the steam table, third column from the last.

The ENTROPY OF VAPORIZATION, n_e, is the CHANGE OF ENTROPY of 1 lb. of fluid which occurs DURING VAPORIZATION. Vaporization takes place at constant temperature. A constant temperature process, in Fig. 21, is represented by a horizontal line. The subtended area is a rectangle. The width of the rectangle is the change of entropy. It is equal to the area divided by the height or to the heat absorbed divided by the absolute temperature. Hence $n_e = L \div (t + 460)$. Values are given in the column of the steam table next to the last.

The TOTAL ENTROPY, OR ENTROPY OF 1 LB. OF VAPOR, is $n_s = n_w + n_e$. Values appear in the last column of the table for steam.

Prob. 137. Compute n_e and n_s for steam at 312° F. (See Prob. 135. Ans., $n_e = 1.1665$, $n_s = 1.62$.)

72. **Entropy Diagram for Steam.** In Fig. 22, the ordinate scale is converted into FAHRENHEIT temperature. As in Fig. 21, the lower part of the diagram is omitted. All heat areas are to be understood as carried down to − 460° F. Water is to be heated from 32°. The state is a, taken on the vertical axis. The path ac represents the heating of water from 32° F. to 150° F. The abscissa of this path (length ot) is $(n_w)_{150} = 0.2149$. The water is then further heated to 327.8° F. along cb. The abscissa of the WHOLE path ab is $os = (n_w)_{327.8} = 0.4743$. The gain of entropy in heating water from 150° to 327.8° is $os - o = ts = 0.4743 - 0.2149 = 0.2594$. The whole curve ab may thus be plotted from the steam table as accurately and as completely as may be desired.

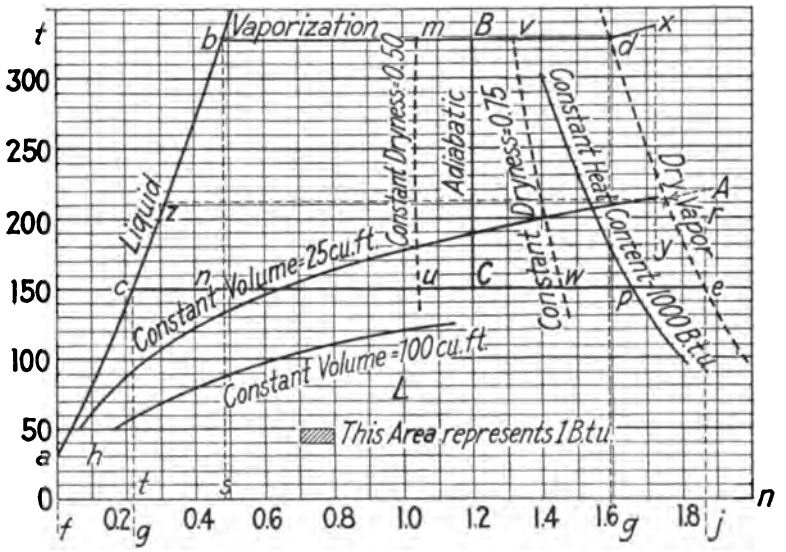

FIG. 22. Temperature-Entropy Diagram for Steam.

Prob. 138. What is the abscissa of that part of the curve *ab* which terminates at 212° F.? (Ans., 0.3118.) How much entropy is gained when 1 lb. of water is heated from 212° to 312°? (Ans., 0.1417.)

Suppose the pressure to be such that the water begins to boil at the point *b*, Fig. 22. Here the temperature is 327.8° F. Vaporization takes place at constant temperature, and is therefore represented by the horizontal line from *b*. At some point *d*, on this line, vaporization will be completed. The length of the line *bd* represents the change of entropy during vaporization, or briefly the ENTROPY OF VAPORIZATION. The symbol for this quantity is n_e. Its value for the assumed condition is

$$(n_e)_{327.8} = bd = (L/T)_{327.8} = 1.1277.$$

If the pressure had been different, so that vaporization had taken place at 150° F., the path representing vaporization would have been *ce*. The dotted line *de* joining the end-points of vaporization paths like *ce* and *bd* is a locus of completely vapor-

ized states. It represents every possible kind of dry steam. Its abscissa at any point is n_s. The curves *ab* and *ed* meet at the CRITICAL TEMPERATURE.

All heat quantities are measured above 32° F. as a starting point. The area under *ac* (*acgf*, carried down to the − 460° F. level) therefore represents the heat of the liquid at *c*, abbreviated h_c. The area under *ce* (*cejg*) similarly carried down, represents the heat consumed in converting liquid at *c* into dry vapor at *e*, abbreviated as L_{ce}. The sum of these two areas represents H_e, the total heat of dry steam at *e*, above 32° F.

Prob. 139. Locate *d*, where $t = 327.8$.

(Ans., $T = 787.8$, $n = (n_s)_{327.8} = 1.602$.)

Any point on *ab* represents boiling water. Any point on *de* represents completely evaporated ("dry") steam. Any point BETWEEN these two curves represents a mixture of steam and boiling water, i.e., WET STEAM. If the point is close to *de*, the mixture is nearly dry steam. If the point is close to *ab*, the mixture is mostly water. If the state moves from a point near *ab* toward *de*, water is being evaporated. If the movement is in the reverse direction, steam is being condensed.

An adiabatic process is represented by a vertical line. If hot water (as at *b*) expands to a lower pressure, adiabatically (path *bn*), some of it will be vaporized, because the point *n* is farther from *ab* than is the point *b*. This actually occurs when the blow-off cock of a boiler is opened. If dry steam is expanded adiabatically (path *dp*), some of it will condense. If very wet steam is compressed adiabatically (path *nb*) it will be partially or wholly liquefied. If nearly dry steam is sufficiently compressed adiabatically (path *pd*) it will be made wholly dry. A point *m*, half way between *ab* and *de*, represents steam "50 per cent. dry," i.e., it represents a mixture in which half the water has been evaporated while the other half is still liquid.

Prob. 140. Locate the point *u*, Fig. 22, representing steam 50 per cent. dry at 150° F. (Ans., entropy at the point $c = 0.2149$, entropy at the point $e = 1.8674$, $n_u = 0.2149 + \frac{1}{2}(1.8674 - 0.2149) = 1.0412$.)

The "constant dryness" curves *mu*, *vw* may be plotted by computations made as in Prob. 140, or may be obtained graphically by properly dividing horizontal distances between *ab* and *de*. Thus $bm = md$, $mv = vd$, $cu = ue$, $uw = we$.

Suppose heat to be added when the state is *d*. There being no liquid present, the temperature will rise; i.e., the path from *d* will be UPWARD. Since heat is supplied, there must be a subtended area. Hence the path will NOT be vertical. Since the sign of the heat supplied is +, the path must be FROM left TO right: i.e., it will slope, so to speak, toward the northeast. If the specific heat during this operation were constant, the path would be a logarithmic curve, indicated by Equation (39), like *dx*. The operation *dx* is one of SUPERHEATING. Superheating implies the addition of heat (no liquid being present), increase of temperature and increase of entropy. All points lying to the right of *de* represent superheated vapor. If superheated steam expands adiabatically (path *xy*) it becomes less superheated, then dry and finally wet. If dry steam (or nearly dry steam) is compressed adiabatically (path *er*) it is superheated. The heat absorbed during the superheating operation *dx* is represented by the area under *dx*, carried down to the − 460° F. level.

73. **Factor of Evaporation.** In usual practice, we do not start the formation of steam with water at 32°, but at some higher temperature, t_0. The heat expenditure for warming from this temperature to t_1 is $h_1 - h_0$ B.t.u. The total expenditure of heat in forming dry steam at $t_1°$ is then $h_1 - h_0 + L_1 = H_1 - h_0$ B.t.u. This varies with the pressure, p_1, at which evaporation occurs, and with the temperature t_0 at which water is supplied in the first place. This temperature is called the FEED-WATER TEMPERATURE. The generation of 1 lb. of dry steam from feed water therefore represents a variable quantity of heat.

For the purpose of comparison, the standard condition to which all others are referred is thus defined: the feed temperature is 212° F. Hence $h_0 = 180$. Also, the steam pressure is normal atmospheric pressure. Hence h_1 is also 180. Then the total

heat expenditure per lb. of steam formed is $h_1 - h_0 + L_1 = L_{212} = 970.4$. This condition is described as FROM (a feed water temperature of) AND AT (a steam pressure corresponding with) 212° F. In Fig. 22, the area under zA represents L_{212}, the heat expenditure necessary to generate 1 lb. of steam "from and at 212° F." The area under cbd represents the heat expenditure when the feed temperature is t_c and the steam pressure is the pressure which corresponds with the temperature t_{bd}. The quotient of the latter area by the former is called the FACTOR OF EVAPORATION, which has the value

$$F = \frac{h_b - h_c + L_{bd}}{970.4} = \frac{H_d - h_c}{970.4}. \tag{40}$$

The factor of evaporation is the ratio of the heat actually employed to make steam, divided by the heat necessary from and at 212° F. A high value of F means costly steam.

Some Equivalents

Mechanical Units	Heat Units
1 ft. lb.	= 1/778 B.t.u.
778 ft. lb.	= 1 B.t.u.
754 971 ft. lb.	= 1 lb. of water evaporated from and at 212° F. = 970.4 B.t.u.
33 000 ft. lb. per min.	= 1 hp. = 42.42 B.t.u. per min.
1 980 000 ft. lb. per hr.	= 1 hp. = 2545 B.t.u. per hr.

Prob. 141. How many B.t.u. are required to heat 10 lb. of water from 212° F. to 312° F.? (Ans., 1020 B.t.u.)

Prob. 142. How many B.t.u. are required to generate 1 lb. of dry steam at 80 lb. pressure from feed water at 90° F.? (Ans., 1124.3.) What is the factor of evaporation? (Ans., 1.16.)

Prob. 143. Find F when $t_0 = 80$, $p_1 = 200$. (Ans., 1.18.)

Prob. 144. A boiler evaporates 10 lb. of water per lb. of coal under the conditions of Prob. 142. Another boiler, using the same coal, evaporates 9.9 lb. of water per lb. of coal under the conditions of Prob. 143. Which is the better boiler? How much better? (Ans., the second boiler is 1.27 per cent. better.)

Wet Vapor: Vapor Processes

74. **Properties of Wet Vapor.** If 1 lb. of a liquid is brought to the boiling point t and the pressure p and then the portion x lb. of this liquid is evaporated, the properties are:

h, B.t.u., heat of liquid; for all of the liquid was brought to the boiling point:

n_w, entropy of liquid; for the substance moved along ab, Fig. 22, in the usual way:

t, the common Fahrenheit temperature of the liquid and the vapor:

xL, the heat of vaporization; for only x lb. were vaporized:

xn_e, the entropy of vaporization; for $xL/T = xn_e$:

$h + xL$, the total heat, which is the sum of the heats of liquid and of vaporization:

$n_w + xn_e$, the total entropy, which is the sum of the entropies of liquid and of vaporization.

The expression for specific volume of wet vapor is less obvious. The volume of 1 lb. of the liquid at the boiling point is v_w. In a MIXTURE weighing 1 lb., $(1 - x)$ lb. of liquid are present. There are also present x lb. of dry vapor, the total volume of which is xv. Then the volume of 1 lb. of the mixture is $(1 - x)v_w + xv = v_w + x(v - v_w)$. This is very nearly equal to xv, and may be so taken unless special note is made to the contrary.

The external work of vaporization was shown to be

$$W = \frac{144p}{778}(v - v_w) \text{ B.t.u.}$$

when DRY vapor was formed. The quantity in the parenthesis is the INCREASE of volume from liquid to dry vapor. The corresponding increase for WET vapor is $x(v - v_w)$. The value of W for vaporization to wet vapor is consequently

$$\frac{144p}{778}x(v - v_w)$$

which is x times the value for dry vapor. The internal work (E effect) in the vaporization to dry vapor was $r = L - W$. For wet vapor it is $xL - xW = x(L - W) = xr$.

Recapitulation:

Properties that are the same for wet vapor as for dry } h, n_w, p, t,

Properties multiplied by x { **dry vapor,** L, n_e, r,
wet vapor, xL, xn_e, xr,

Other properties { **dry vapor,** v, H, n_s,
wet vapor, $v_w + x(v - v_w)$, $h + xL$, $n_w + xn_e$,
(approximately xv**).**

The quantity x is the DRYNESS FRACTION or PROPORTION OF DRYNESS. It is the weight of dry vapor in 1 lb. of the mixture of dry vapor and liquid.

Prob. 145. From values given in the steam table at 100 lb. pressure, compute all of the corresponding properties for steam at this pressure if 0.90 dry. (Ans., $p = 100$, $t = 327.8$, $h = 298.3$, $n_w = 0.4743$, $xL = 799.2$, $xn_e = 1.0149$, $xr = 725.94$, $v_w + x(v - v_w) = 3.9876$ if $v_w = 0.0177$, $h + xL = 1097.5$, $n_w + xn_e = 1.4892$.)

The factor of evaporation for a wet vapor, following Equation (40), is

$$F_w = \frac{xL + h - h_0}{970.4}, \tag{41}$$

where quantities without subscripts refer to the vapor condition and h_0 to the feed water condition.

Prob. 146. What is the factor of evaporation for the steam in Prob. 145, if that steam was generated from feed water at 90° F.? (Ans., 1.07.)

75. **Volumes on the *Tn* Diagram.** Suppose it to be required that we plot in Fig. 22 the locus of states where the volume is 25 cu. ft. Write

$$25 = v_w + x(v - v_w), \text{ or briefly, } xv.$$

The values of v_w in the following are from Fig. 22*a*. Those of v, n_w and n_e are from the steam table:

For t	50	75	100	125	150	175	200	215
v =	.1702	743	350.8	178.4	96.9	55.7	33.6	25.35
v_w =	0.01602	0.01603	001603	0.01623	0.01634	0.01648	0.01663	0.01673
$x = \frac{25-v_w}{v-v_w}$ =	0.015	0.034	0.071	0.14	0.258	0.45	0.743	0.987
n_e =	2.0865	1.9631	1.8505	1.7475	1.6525	1.5645	1.4824	1.4354
xn_e =	0.031	0.0661	0.132	0.245	0.426	0.702	1.103	1.42
n_w =	0.0361	0.0840	0.1295	0.1730	0.2149	0.255	0.2937	0.3163
n_w+xn_e =	0.0671	0.1501	0.2615	0.4180	0.6409	0.957	1.3967	1.7363

Values in the last line, plotted against those in the first line, give the CONSTANT VOLUME curve of 25 cu. ft. in Fig. 22. Curves

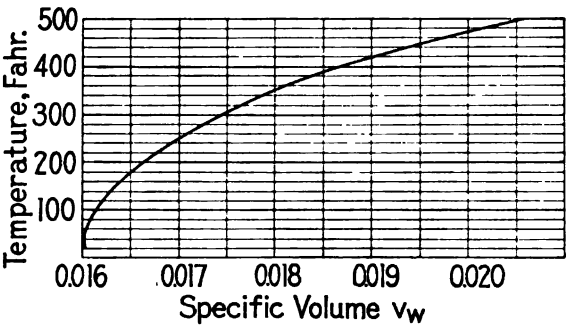

FIG. 22*a*. Specific Volume of Water.

for less than 25 cu. ft. lie above this: curves for larger volumes lie below it. The maximum volume on the diagram is toward the "south-east" corner.

Prob. 147. Find the entropy of steam at 50° F. when its volume is 100 cu. ft. (Ans., 0.158.) Find also at 75°, 100°, 125°. (Ans., 0.349, 0.658, 1.152.)

76. **Constant Heat Content.** The total heat of wet steam is $h + xL$. This may have any value between rather wide limits, depending on p and x. Thus it may be 1000 B.t.u. when $p = 100$ if $x = (1000 - h)/L = 701.7/888 = 0.791$. Then $n_w + xn_e = 0.4743 + (0.791 \times 1.1277) = 1.366$. From values found in this

* Note that it is sufficiently accurate to take $x = 25/v$. This would not be a safe procedure, however, if we were plotting a line of very small volume.

way, the curve of constant heat content = 1000 B.t.u. is plotted in Fig. 22.

Prob. 148. What are the dryness and entropy of steam at atmospheric pressure when the heat content is 1000 B.t.u.? (Ans., $x = 0.845$, $n_w + xn_e = 1.53$.)

Prob. 149. If the steam in Prob. 148 emits 500 B.t.u. without change of pressure, what is then its dryness? (Ans., 0.33.) If it then emits 406 B.t.u. more, what does it become? (Ans., water at 126.15° F., if the pressure is allowed to fall after condensation begins: within a small fraction of 1° of this temperature if the pressure is kept at that of the atmosphere.)

Prob. 150. A boiler generates from feed water at 90°, 20 000 lb. of steam per hr. at 200 lb. pressure, 0.99 dry. The coal has a heat value of 11 316.7 B.t.u. per lb. and the efficiency of the boiler is 0.75. What weight of coal is burned per hr.? (Ans., 2667 lb.)

77. **Work Done in Heating Liquid.** The heat absorbed in the operation *ab*, Fig. 22, is h_b per lb. of liquid. The external work done during this heating, and included in h_b, is very small and may often be neglected. The method of computing it should be known.

A pound of water has been heated from 32° F. and a pressure of 0.0886 lb. per sq. in. (Art. 63), until its pressure is that corresponding with t_b, or 100 lb. per sq. in. The external work is that due to this increase of pressure. It is equivalent to the work of LIFTING the pound of water to a height represented by the difference of the final and initial pressures.* What would be this height? Suppose the water to be delivered into the base of a vertical tube of 1 sq. ft. area. If the height to the top of the

*When we lift water 255 ft., its pressure (as measured on a gage placed at the original level) is approximately $\frac{14.7}{34.0} \times 255 = 110$ lb., if the temperature is around 60°. (Atmospheric pressure is equivalent to a head of 34 ft. of water.) Conversely, the application of 110 lb. pressure is equivalent to lifting the water 255 ft. The more complicated but more correct statement in the text is due to the fact that the water in question is at a temperature of 327° after it is lifted and at a temperature of 32° before lifting occurs.

tube (where the water flows off) is f ft., and the density of the water is d, the hydrostatic pressure at the base of the tube is simply the weight of water, or fd lb. per sq. ft. Or, inversely, the height in ft. is the pressure per sq. ft. divided by the density. At 100 lb. pressure per sq. in., the density of water is 56.4 lb. per cu. ft. The external work of our process is equivalent to lifting 1 lb. of water to a height of (substantially) $100 \times 144 \div 56.4 = 255$ ft., or is 255 ft. lb. In general symbols,

$$\text{External Work} = W_h = \frac{144 \times pv_w}{778} = \frac{pv_w}{5.403} \text{ B.t.u.}, \quad (42)$$

for v_w is the reciprocal of the density. The gain of internal energy during the heating of the liquid is $h - (pv_w/5.403)$. The external work done in heating 1 lb. of liquid from c to b is $(W_h)_b - (W_h)_c$. Values of v_w are given in Fig. 22a.

Prob. 151. Find W_h at the critical temperature, where $v_w = 0.049$. (Ans., 26.7 B.t.u.)

78. **Vaporization or Condensation.** As shown in rAt. 74, the heat absorbed in partial vaporization is xL B.t.u. per lb. For complete vaporization, put $x = 1.0$. Vaporization (partial) at constant pressure is represented by such a line as bv, Fig. 22. CONDENSATION of steam initially at the point v would be represented by the reverse line vb. The heat EMITTED is $(xL)_v$, and its sign is NEGATIVE. Note that the isothermal and the constant pressure paths coincide when the fluid consists of a mixture of liquid and vapor. The external work done (Art. 74) is $144\ px(v - v_w) \div 778$ B.t.u. This is + for vaporization and $x = 1$ if vaporization is complete. It is − for condensation. The internal work is $+ xr$ for vaporization and $- xr$ for condensation.

Prob. 152. How many B.t.u. are emitted when 1000 lb. of steam at 1 lb. pressure, 0.70 dry, are condensed and cooled to 90°? (Ans., 736 020 B.t.u.) What is the EXACT loss of internal energy? Take $v_w = 0.01614$ at 1 lb. and 0.0161 at 90°. (Ans., 692 829 B.t.u.)

79. **Adiabatic Process.** No heat is absorbed or emitted if the process is adiabatic. The external work is computed by writing

as the law of a vapor adiabatic, initial entropy = final entropy: or, (Fig. 22)

$$n_B = n_C,$$
$$(n_w + xn_s)_B = (n_w + xn_s)_C. \tag{43}$$

This is one of the most useful equations of our subject. Either value of x may be 1.0. For an adiabatic,

$$H = 0 = E + W, \qquad W = -E = E_B - E_C,$$
$$W = \left(h - \frac{pv_w}{5.403} + xr\right)_B - \left(h - \frac{pv_w}{5.403} + xr\right)_C. \tag{44}$$

This is very nearly equal to $(h + xr)_B - (h + xr)_C$, and in ordinary calculations should be so taken. Do not use this approximation, however, in Probs. 153–156.

The work is + when the path is DOWNWARD and − when it is UPWARD.

On a pv diagram, the vapor adiabatic is a smooth curve which may be represented approximately by the equation $p_B v_B^a = p_C v_C^a$, but the value of a depends on the initial dryness: $a = 1.035 + 0.1x_B$. Then $v_C = v_B(p_B/p_C)^{1/a}$ and

$$W = \frac{P_B v_B - P_C v_C}{a - 1} = 144\left(\frac{p_B v_B - p_C v_C}{a - 1}\right)\text{ft. lb.}$$

Prob. 153. Steam at 250 lb. pressure, 0.99 dry, expands adiabatically to 1 lb. pressure. What is the dryness after expansion? (Ans., 0.7515.) What was the specific volume before expansion? (Ans., 1.832 if $v_w = 0.0187$.) After expansion? (Ans., 250.)

Prob. 154. In Prob. 153, find $pv_w/5.403$ at the initial condition. (Ans., 0.87 B.t.u.) At the final condition, when $v_w = 0.01614$. (Ans., 0.003 B.t.u.) What was the internal energy before expansion? (Ans., 1108.91.) After expansion? (Ans., 800.93.) What was the external work done during expansion, by Equation (44)? (Ans., 307.98 B.t.u.)

Prob. 155. What is the value of a in the exponential approximate formula for this process? (Ans., 1.134.) Of v_C? (Ans., 239.) Of the external work? (Ans., 302 B.t.u.)

Prob. 156. Consider the adiabatic expansion of Prob. 153 as *BC*, Fig. 22. How much heat was absorbed, and how much external work was done, during the operation *cb*? (Ans., heat = 305.4 B.t.u., work = 0.867 B.t.u.) Along *bB*? (Ans., heat = 818.04 B.t.u., work = 83.46 B.t.u.) Along *Cc*? (Ans., heat = − 777.5 B.t.u., work = − 46.37 B.t.u.) Show that for the

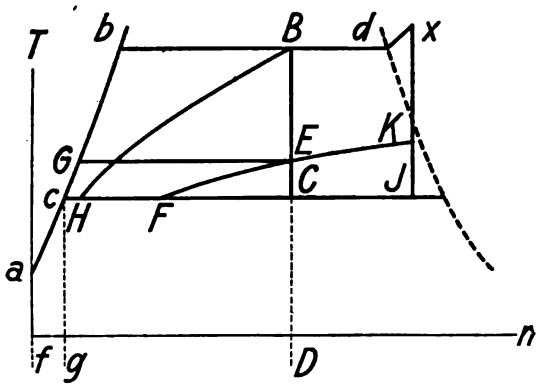

FIG. 22*B*. Vapor Paths and Cycles.

closed figure *cbBC*, $\Sigma H = \Sigma W$. (Ans., $\Sigma H = 305.4 + 818.04 - 777.5 = 345.94$. Also $\Sigma W = 0.867 + 83.46 + 307.98 - 46.37 = 345.94$ B.t.u.)

Prob. 157. Ammonia is compressed adiabatically from 0° to 90°. At 0°, $n_w = -0.0709$, $n_s = 1.245$. At 90°, $n_s = 1.0219$. What must be the initial dryness if the fluid is to be exactly dry ($x = 1.0$) after compression? (Ans., 0.878.)

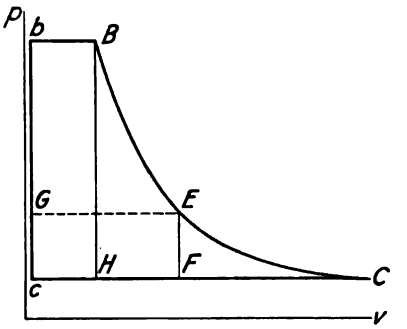

FIG. 23. Ideal Cycles for Vapors.

80. **Ideal Vapor Cycle, Complete Expansion: "Rankine" Cycle.** The cycle *cbBC*, Fig. 23, represents the ideal operation of a STEAM POWER PLANT running condensing. Water is

heated along *cb*, the pressure and temperature increasing and the volume increasing slightly. Vaporization occurs along *bB*, adiabatic expansion along *BC* and condensation along *Cc*. The corresponding diagram is similarly lettered in Fig. 22*B*. We have seen how to compute the work and heat for each path of this cycle. The following is a briefer method of computing efficiency and mean effective pressure. Referring to Fig. 22*B*, the efficiency is

$$\begin{aligned} e &= cbBC \div gcbBD \\ &= (fabBD - facCD) \div (fabBD - facg) \\ &= (H_B - H_C) \div (H_B - h_c), \end{aligned}$$

where H indicates the total heat above 32° of steam EITHER WET OR DRY. This is best written in the specific form,

$$e = \{(h + xL)_1 - (h + xL)_2\} \div \{(h + xL)_1 - h_2\}, \quad (45)$$

in which subscripts $_1$ and $_2$ refer to high-pressure and low-pressure conditions, respectively. This efficiency is always less than that of the Carnot cycle between the same extreme temperatures. The value of x_1 is usually close to 1.0. That of x_2 is computed from Equation (43): $(n_w + xn_e)_1 = (n_w + xn_e)_2$.

The mean effective pressure is the work in ft. lb. divided by $\{144 \times (v_C - v_c)\}$, Fig. 23. This divisor becomes $x_2(v_2 - v_c)$, or with an error practically never exceeding 1/10 of 1 per cent. for steam, simply x_2v_2, where v_2 is the tabular volume of dry steam at the low-pressure condition. Then

$$p_m = \frac{(h + xL)_1 - (h + xL)_2}{x_2v_2} \times 5.403. \quad (46)$$

Prob. 158. Find the efficiency of a Rankine steam cycle in which the extreme pressures are 250 and 1 lb. and the dryness at the beginning of expansion is 0.99. (Ans., 0.308.) How much net work is done? (Ans., 345.94 B.t.u.: compare Prob. 156.) What is the mean effective pressure? (Ans., 7.5 lb.: mean effective pressures are very low in this type of cycle.)

81. **Incomplete Expansion.** During the later stages of expansion, Fig. 23, very little work is derived from the fluid in pro-

portion to the increased displacement. The cylinder of the engine will therefore be large and costly for its power. (This does not apply to turbines.) In usual engine practice, expansion is terminated at some point E, before the pressure has been reduced to that at which the steam is to be condensed. A constant-volume DROP, EF, then occurs. See Fig. 22B also. The cycle is $cbBEF$. Divide it into two areas $GbBE$ and $GEFc$ by a horizontal line. The lower area is on the pv diagram very nearly a rectangle, and its magnitude in B.t.u. may be taken as

$$\frac{144}{778} v_E(p_E - p_F).$$

Then the efficiency is

$$e = \frac{\Sigma W}{(h + xL)_1 - h_2}$$

$$= \frac{(h + xL)_1 - (h + xL)_E + \frac{144}{778} v_E(p_E - p_2)}{(h + xL)_1 - h_2}, \quad (47)$$

where v_E is the volume of wet vapor at the point E, to be computed by first finding the entropy and dryness.

Thus under the limiting pressures and initial dryness of Prob. 158, let $p_E = 10$. Then at this pressure $n_w = 0.2832$, $n_e = 1.5042$. Equating the entropies at B and E,

$$0.5676 + (0.99 \times 0.9600) = 0.2832 + 1.5042x_E,$$

$$x_E = 0.819.$$

Then, very nearly, $v_E = 0.819 \times 38.38 = 31.4$, the value 38.38 being that for dry steam at 10 lb. pressure. Then the area $GEFc = (144/778) \times 31.4 \times 9 = 52.1$ B.t.u. The value of $(h + xL)_E$ is $161.1 + (0.819 \times 982) = 967$ B.t.u. The area $bBEG$ is $375.2 + (0.99 \times 826.3) - 967 = 226.24$ B.t.u. Then $\Sigma W = 52.1 + 226.24 = 278.34$ B.t.u. and $e = 278.34 \div (1193.24 - 69.8) = 0.247$. This is less than the result in Prob. 158. If expansion had terminated sooner, i.e., if p_E had been greater than 10, the efficiency would have been still less.

Prob. 159. Find the efficiency as above, when $p_E = 20$.

(Ans., 0.215.)

Prob. 160. Find the efficiency as above in the limiting case, when $p_E = 250$, cycle $cbBH$. (Ans., 0.074.)

The mean effective pressure with the cycle $cbBEF$ is, very nearly, $778\Sigma W \div 144v_E$, or

$$p_m = 5.403\frac{(h + xL)_1 - (h + xL)_E}{v_E} + p_E - p_2 = \frac{5.403\Sigma W}{v_E}. \quad (48)$$

For the foregoing conditions, this becomes 48.1. It may be as high as desired, up to the value $p_1 - p_2$. Incomplete expansion reduces efficiency, but increases mean effective pressure.

Prob. 161. Find p_m in Probs. 159, 160. (Ans., 78 and 249 lb.)

82. **Ideal Steam Consumption.** The area of the cycle, or the numerator of the efficiency expression, represents the ideal work obtainable per lb. of steam, ΣW (B.t.u.). Since 1 hp. = (33 000 $\times$ 60) $\div$ 778 = 2545 B.t.u. per hr., the IDEAL STEAM RATE (lb. of steam per hr. per Ihp. under best conditions) is $2545 \div \Sigma W$. For Prob. 158, this rate is $2545 \div 345.94 = 7.33$ lb. The heat supplied being $h_1 + x_1L_1 - h_2$, an engine having an efficiency of 100 per cent. would use $2545 \div (h_1 + x_1L_1 - h_2)$ lb. of steam per hr. per Ihp. For the conditions of Prob. 158, this is $2545 \div 1123.44 = 2.26$ lb. But this ideal engine has an efficiency of only 0.308. Hence its ideal steam rate is $2.26 \div 0.308 = 7.33$ lb., as before. The actual steam consumption will be greater than 7.33, under the assigned conditions.

The IDEAL HEAT RATE is the HEAT CONSUMED IN B.T.U. PER MIN. PER IHP. At 100 per cent. efficiency, it would be 33 000 $\div$ 778 = 42.42. At an efficiency e it is $42.42 \div e$. In Prob. 158, it is $42.42 \div 0.308 = 138$ B.t.u.

Prob. 162. Find ideal steam rate and ideal heat rate in Art. 81. (Ans., 9.13 lb., 171 B.t.u.)

Prob. 163. Find these ideal rates for Prob. 160.
(Ans., 30.2 lb., 570 B.t.u.)

Prob. 164. Find the ideal rates in Prob. 159.
(Ans., 10.4 lb., 195 B.t.u.)

Fig. 24 shows the variation in efficiency of such cycles as *cbBEF* for steam initially dry under the initial (p_B) and back (p_F) pressures common in condensing and non-condensing practice.

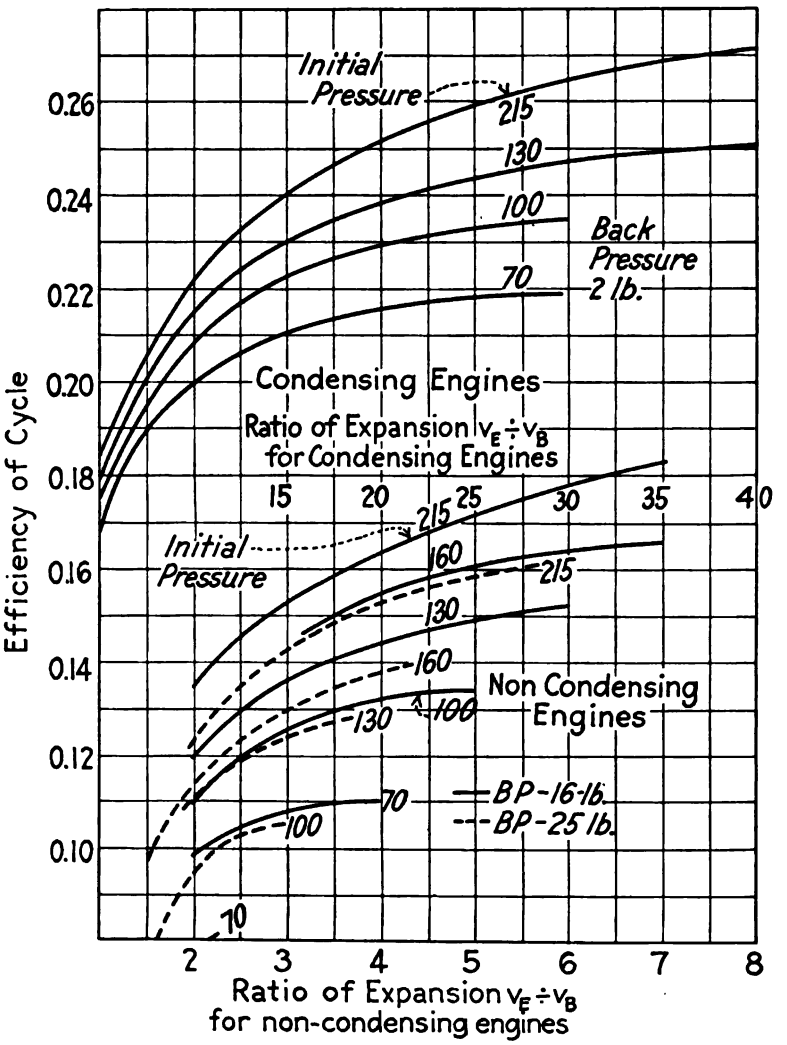

FIG. 24. Ideal Efficiencies with Incomplete Expansion: Steam Initially Dry.

Prob. 165. From Fig. 24, state the heat rate and steam rate for an ideal cycle using dry steam at 215 lb. pressure, condensing at 2 lb. absolute pressure, when the ratio of volumes during adiabatic expansion is 30. (Ans., $e = 0.265$, heat rate = 160, steam rate = 8.7.)

Superheated Vapor

83. **Properties.** Superheating almost always occurs at constant pressure. The pressure is that at which the saturated vapor was formed. Vapor must be dry-saturated before it can be superheated. The specific heat at constant pressure is highly variable, both with pressure and with temperature. Extensive tables are therefore necessary for mean values of k, the specific heat, or of H', the total heat. If t = saturation temperature, t' = actual vapor temperature, both at the pressure p, the "SUPERHEAT" (amount or number of degrees of superheat) is $t' - t$ and the total heat is

$$H' = H + k(t' - t).$$

The volume is given quite closely for steam by the rather formidable equation

$$pv' = 0.5962(t' + 460) - p(1 + 0.0014p)\left(\frac{150\,300\,000}{(t' + 460)^3} - 0.0833\right).$$

The gain of entropy of a vapor during superheating may be expressed as $k \log_e (t' + 460)/(t + 460)$, and the total entropy is

$$n' = n_s + k \log_e \frac{T'}{T}.$$

The external work done during superheating is

$$W' = \frac{144p}{778}(v' - v) \text{ B.t.u.}$$

The whole amount of external work done during the formation of superheated steam from water at 32° and 0.0886 lb. pressure is (see Arts. 77, 78),

$$\frac{pv_w}{5.403} + \frac{p(v - v_w)}{5.403} + \frac{p(v' - v)}{5.403} = \frac{pv'}{5.403} \text{ B.t.u.}$$

The gain of internal energy during this operation is then

$$H' - \frac{pv'}{5.403}.$$

Prob. 166. For steam at 250 lb. pressure, superheated 200°, $k = 0.562$. Find t', H', v', n', W' and the gain of internal energy from liquid at 32°. (Ans., 601.1° F.: $H' = 1313.9$: $v' = 2.471$: $n' = 1.646$, $W' = 28.7$, gain of energy = 1199.7.)

The factor of evaporation for superheated steam is

$$F' = (H' - h_0) \div 970.4,$$

following Equation (40).

Prob. 167. What is the factor of evaporation for the steam in Prob. 166, if formed from water at 90°? (Ans., 1.29.)

The following is a greatly abridged sample of a superheated steam table.

Properties of Superheated Steam

(Condensed from STEAM TABLES AND DIAGRAMS, by Marks and Davis, with the permission of the publishers, Messrs. Longmans, Green, & Co.)

SUPERHEAT, °F	40	90	200	300
Absolute Pressure Lbs. per Square Inch	t' = 141.7	191.7	301.7	401.7
	v' = 357.8	387.9	453.7	513.4
1	H' = 1122.6	1145.3	1195.6	1241.5
	n' = 2.0069	2.0434	2.1145	2.1701
	t' = 367.8	417.8	527.8	627.8
	v' = 4.72	5.07	5.80	6.44
100	H' = 1208.4	1234.6	1289.4	1337.8
	n' = 1.6294	1.6600	1.7188	1.7656
	t' = 393.1	443.1	553.1	653.1
	v' = 3.44	3.70	4.24	4.71
140	H' = 1215.8	1242.8	1298.2	1346.9
	n' = 1.6031	1.6338	1.6916	1.7376
	t' = 398.5	448.5	558.5	658.5
	v' = 3.22	3.46	3.97	4.41
150	H' = 1217.3	1244.4	1300.0	1348.8
	n' = 1.5978	1.6286	1.6862	1.7320
	t' = 441.0	491.0	601.0	701.0
	v' = 1.98	2.14	2.47	2.75
250	H' = 1229.3	1257.7	1313.9	1363.5
	n' = 1.5593	1.5900	1.6460	1.6905

84. **Rankine Cycle with Superheat.** Such a cycle is represented as *cbdxJ*, Fig. 22*B*, or as *cbBC*, Fig. 23. There is no change in direction of the constant pressure line on the *pv* diagram. The efficiency is

$$e = \frac{(H')_x - (h + xL)_J}{(H')_x - h_c}. \tag{49}$$

Thus, take the conditions as $p_x = 250$, $t_x = 601°$, $p_c = 1$. First find the dryness at J:

$$(n_w + xn_e)_J = n' = 1.646 \qquad \text{(Prob. 166)}$$
$$0.1327 + 1.8427x_J = 1.646, \qquad x_J = 0.82.$$

Taking from Prob. 166 $H_x' = 1313.9$,

$$e = \frac{1313.9 - \{69.8 + (0.82 \times 1034.6)\}}{1313.9 - 69.8} = 0.319.$$

In Prob. 158, the efficiency for the same pressures but with SLIGHTLY WET steam was 0.308. SUPERHEATING IMPROVES EFFICIENCY.

The mean effective pressure, following Equation (46), is

$$p_m = \frac{5.403\Sigma W}{(xv)_J}.$$

Noting that ΣW is the numerator of the expression for efficiency, this becomes $(5.403 \times 397) \div (0.82 \times 333) = 7.84$ for our conditions.

Prob. 168. Find e and p_m in Art. 84 if the superheat is 100° instead of 200°. (Ans., $e = 0.313$, $p_m = 7.64$.)

85. **Incomplete Expansion.** Following Art. 81, if expansion terminates before the line of back pressure is reached, the work of the superheated cycle is

$$\Sigma W = H_x' - (h + xL)_3 + \frac{v_3(p_3 - p_c)}{5.403},$$

in which subscripts $_3$ refer to the "terminal" condition, E, Fig. 23. The cycle is *cbBEF* (*cbdxKF* in Fig. 22*B*). The efficiency is

$$e = \frac{\Sigma W}{H_x' - h_c} \tag{50}$$

and the mean effective pressure is

$$p_m = \frac{5.403\Sigma W}{v_K}. \tag{51}$$

Suppose that under the conditions of Art. 84 the terminal pressure is $p_K = p_3 = 10$. Then

$$n_x' = 1.646 = (n_w + xn_s)_3 = 0.2832 + 1.5042x_3,$$

$$x_3 = 0.906, \qquad (xL)_3 = 0.906 \times 982 = 891,$$

$$v_K = 0.906 \times 38.38 = 34.9 = v_3,$$

$$\Sigma W = 1313.9 - 161.1 - 891 + \frac{34.9 \times 9}{5.403} = 320 \text{ B.t.u.},$$

$$e = \frac{320}{1313.9 - 69.8} = 0.257, \quad p_m = \frac{5.403 \times 320}{34.9} = 49.6.$$

The efficiency exceeds that computed for saturated steam in Art. 81.

Prob. 169. Find e and p_m when the pressure limits are 250 and 1 lb., the initial superheat 100° and the terminal pressure 20 lb. (Ans., 0.223 and 79 lb.)

86. **Cases Not Hitherto Considered.** In Arts. 84 and 85, the steam at the end of its adiabatic expansion was WET. In cases where at this point it is still superheated, an exact solution is not possible by any method thus far shown. We know in general the terminal pressure and entropy. We have no way of finding terminal volume, temperature, superheat, or heat content. A graphical method for finding these properties will now be developed.

In Fig. 25 the coördinates are total heat above 32° and total entropy. Plot the "saturation curve" ab from values in the steam table. Thus at 100 lb. pressure the coördinates are 1186.3 and 1.602: at 1 lb. they are 1104.4 and 1.9754. These values establish the points b and n.

The points e and c are those at which the pressures are 150 and 50 lb. Plot the CONSTANT PRESSURE curves ef and cd. This is done by finding the coördinates of points like f, for some

assumed dryness. Thus, assume a dryness of 0.80. Then at f the coördinates are, $330.2 + (0.8 \times 863.2) = 1020.8$, and $0.5142 + (0.8 \times 1.055) = 1.3582$. Taking other drynesses, additional points may be located to establish the whole line ef. Plot cd in the same way. Joining f and d gives a line of CONSTANT DRYNESS.

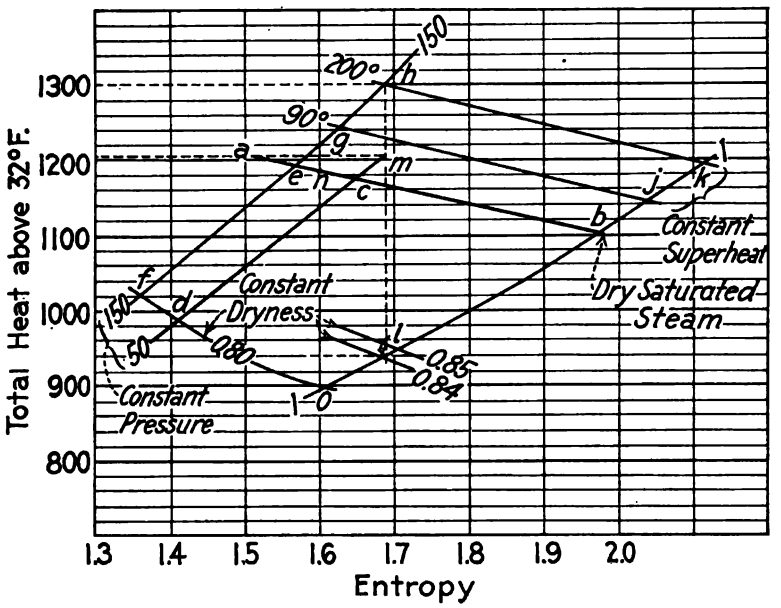

FIG. 25. Development of a Heat Chart.

Prob. 170. Locate the point o on the line of 1 lb. pressure, where the dryness is 0.80. (Ans., coördinates are 897.5 and 1.6069.)

The constant-pressure curves are now carried upward into the superheated region, using the table in Art. 83, and lines of CONSTANT SUPERHEAT, gj, hk, are drawn.

Prob. 171. Find coördinates of points g and j. (Ans., 1244.4 and 1.6286: 1145.3 and 2.0434.)

87. **Application of Total Heat Entropy Diagram.** For COMPLETE EXPANSION (Rankine) cycles, Equations (45) and (49) may both be written

$$e = \frac{Q_1 - Q_2}{Q_1 - h_2}, \tag{52}$$

where subscripts 1 and 2 refer to conditions at the beginning and end of adiabatic expansion, respectively, h stands for heat of liquid and Q for total heat of vapor either wet, dry or superheated. Values of Q_1 and Q_2, and also of the final dryness, x_2, may be read directly from a chart like Fig. 25. A more complete chart appears in Fig. 26. Still larger scale charts, which may be used for accurate work, will be found in recent steam tables like those of Goodenough or Marks and Davis. The saving in labor due to their use is very great. Thus suppose a cycle with complete expansion to work between pressure limits of 150 lb. and 1 lb., with an initial superheat of 200°. Locate h, Fig. 25 and read off $Q_1 = 1300$ at the left. Draw hl vertically. Read off, at l, interpolating between the 0.84 and 0.85 dryness curves, $x_2 = 0.842$, $Q_2 = 944$. Then $e = (1300 - 944) \div (1300 - 69.8) = 0.256$. The value $h_2 = 69.8$ is from the steam table. Taking the specific volume at 1 lb. pressure at 333, also from the steam table,

$$v_2 = 0.842 \times 333 = 280.4, \quad p_m = \frac{5.403 \times 356}{280.4} = 6.86.$$

This method is applicable whether the steam is initially wet, dry or superheated.

Incomplete Expansion. The most difficult case is the one thus far omitted, that in which the steam is superheated at the end of expansion. Equations (47) and (50) may be written

$$e = \frac{Q_1 - Q_3 + \frac{v_3(p_3 - p_2)}{5.403}}{Q_1 - h_2}, \tag{53}$$

where 3 is the TERMINAL condition. If this condition is one of WET steam, the volume v_3 is found from the dryness, as above. But suppose steam at 150 lb. pressure, 200° superheated, to expand to $p_3 = 50$ lb. pressure and then to be condensed at 1 lb. pressure. In Fig. 25, $Q_1 = 1300$, as before: draw hm

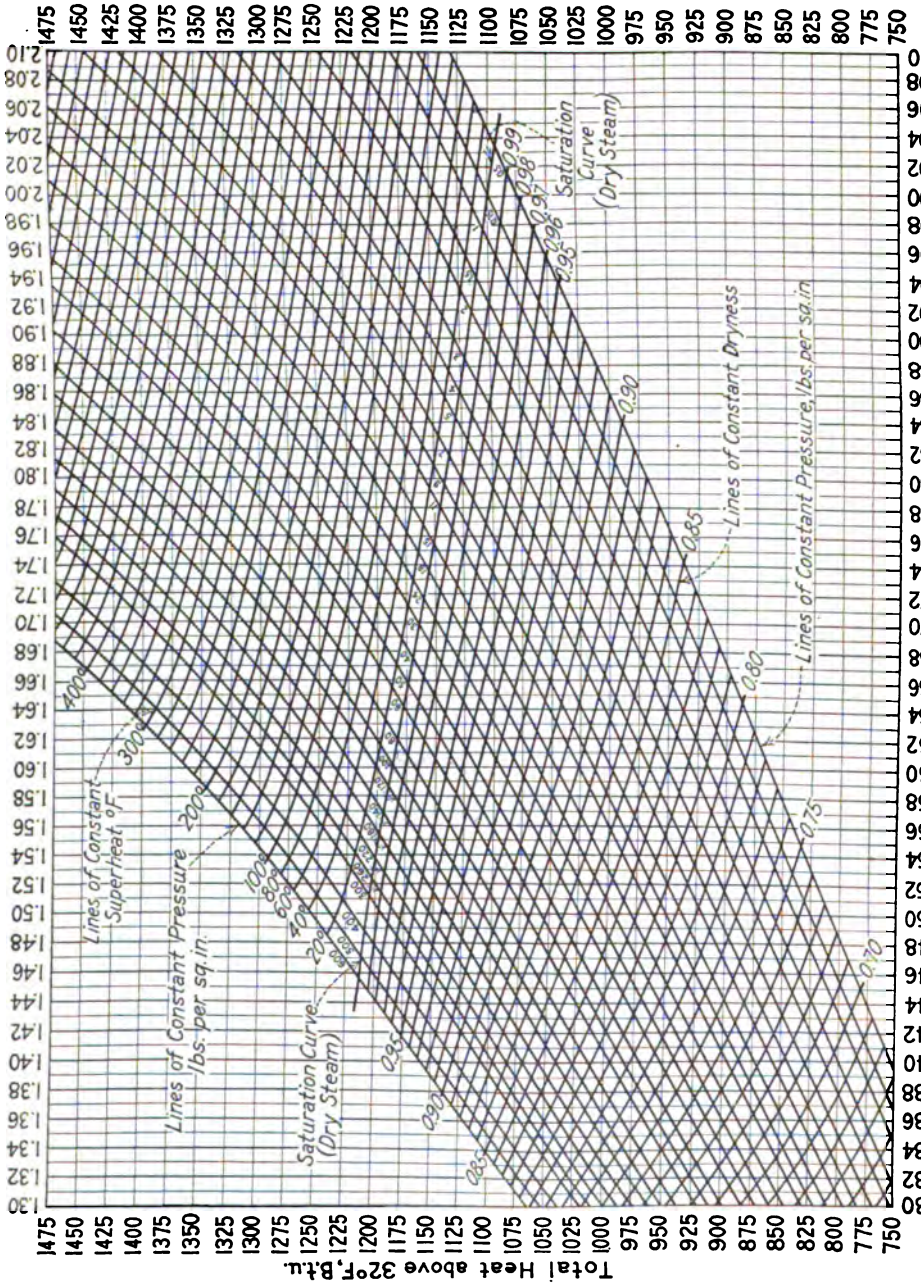

FIG. 26. Total Heat-Entropy Digaram.

vertically. At m, $Q_3 = 1203$, superheat = 54°. Then $t_3 = 281 + 54 = 335$, and the equation in Art. 83 for the volume of superheated steam gives $v_3 = 9.26$. Then

$$e = \frac{1300 - 1203 + \frac{9.26}{5.403}(50 - 1)}{1300 - 69.8} = \frac{181}{1230} = 0.147$$

and

$$p_m = \frac{181 \times 5.403}{9.26} = 106.$$

Note the marked difference between these results and those of the last paragraph. Incomplete expansion has reduced the efficiency from 0.256 to 0.147 but has INCREASED the mean effective pressure from 6.86 to 106. A corresponding increase would occur in the power of an engine of given size and speed.

Prob. 172. Check from Fig. 26 all drynesses and efficiencies computed in Probs. 153, 158–160, 168, 169.

To avoid the labor of computing v_3 from the long formula of Art. 83, an auxiliary chart called the TOTAL HEAT-PRESSURE diagram is often used. This uses the quantities mentioned as coördinates, and plots curves of constant dryness, superheat and volume. Pressure and heat content being taken from Fig. 26, volume may be read directly from the total heat-pressure diagram.

Factors Affecting Ideal Engine Efficiency

88. **Incomplete Expansion, Dry or With Superheat.** Efficiencies of various cycles with superheat are plotted in Fig. 27. These may be compared with values in Fig. 24.

Back Pressure has of course a pronounced influence. This explains the gain in economy by running condensing. A very low ratio of expansion, running condensing, gives better efficiency than a high ratio running non-condensing. A back pressure around 25 lb. absolute is often used for small auxiliary engines on ships. This is for the purpose of using the exhaust steam from these engines advantageously in heating the boiler feed water. Fig. 24 shows that it leads to low engine efficiency. It

pays, because practically all of the heat wasted by the engines goes back to the boilers.

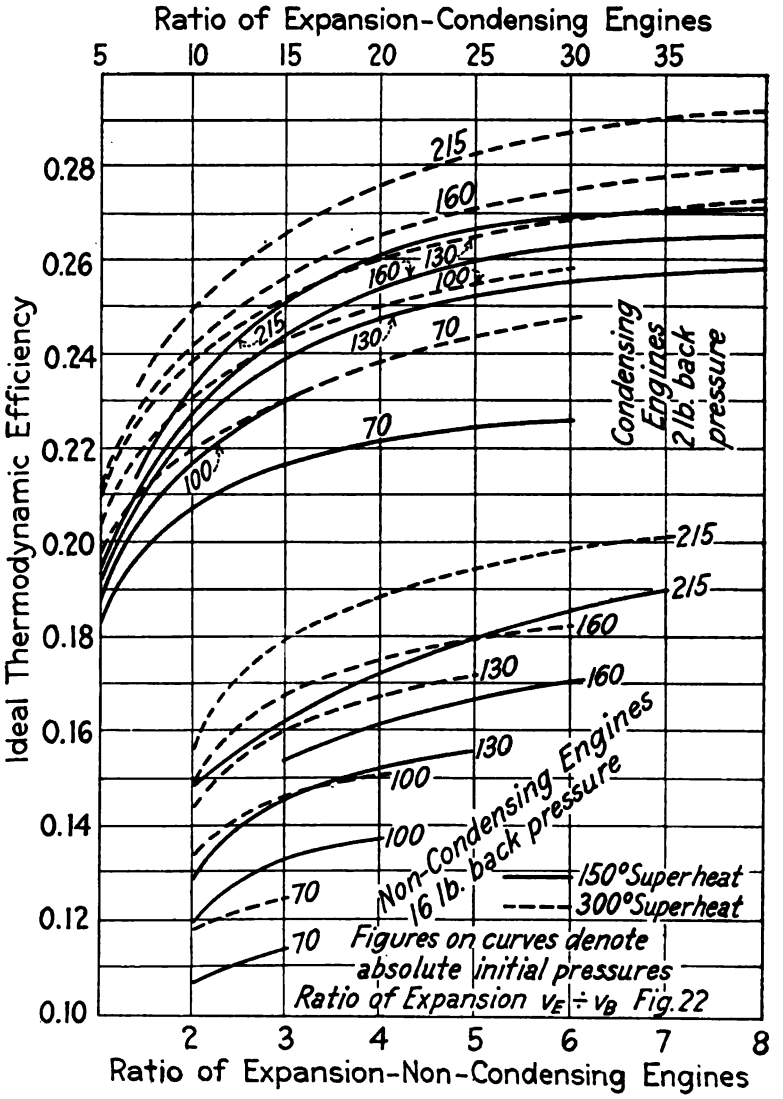

FIG. 27. Ideal Efficiencies with Superheated Steam.

Ratio of Expansion has an especially important bearing on efficiency. Many land engines running at constant speed are governed by varying this ratio. Hence the curves show how efficiency varies with output. The best ratio is not to be determined by thermodynamic considerations alone, but is established from various practical and commercial factors (Arts. 95, 96). At a given pressure, efficiency increases with the ratio of expansion, but mean effective pressure decreases. A ratio is finally reached at which there is little or no further gain in efficiency. This limit is very soon attained if the initial pressure is low, and is postponed if high superheat is used.

High Initial Pressure is important. At a given ratio of expansion, the efficiency increases with the pressure: but for very low ratios, high pressures are of little advantage. In condensing engines with superheat, high pressure is less important than in non-condensing engines using superheat.

Superheat increases efficiency at a given pressure or ratio of expansion. In general, the gain of efficiency is roughly proportional to the amount of superheat. Superheat should not be continued beyond the point where the terminal condition is dry. Superheat gives high efficiency at moderate ratios of expansion and therefore permits of reasonably high mean effective pressures along with good efficiency.

Initial Dryness is not considered in these diagrams. Within reasonable limits (0.95 to 1.0) it has practically no influence on efficiency.

All efficiencies are below those of the Carnot cycle between the same extreme temperature limits.

89. **Vapors Other Than Steam.** In the CARNOT cycle, the efficiency is independent of the fluid used. In the actual steam cycles here described, it depends on the properties of the fluid (latent heat, specific heats). Various fluids have been used in vapor engines. In simple condensing engines with low ratios of expansion, a considerable gain is possible over the best results with steam. A desirable vapor would be one whose pressure

was less than that of steam at high temperatures and more at low temperatures. The latter characteristic would make high vacuums unnecessary. It should have a high latent heat of vaporization and a low specific heat of liquid, since this, in Fig. 22*B*, would make the Rankine cycle approximate more closely the Carnot. In the cycle *cbBC* of this diagram, the heat supplied by the boiler is the area under *cbB*. That carried away at the condenser is the area under *Cc*. The ratios of these areas to the power developed depend on the properties of the fluid used. Hence a suitable substitute fluid might decrease the relative size of boiler or condenser necessary. The mean effective pressure, and hence the size of cylinder, also depends on the properties of the vapor.

Prob. 173. In Fig. 20, what vapor apparently has a higher pressure than that of water at low temperatures, and a lower pressure at very high temperatures?

Mixtures of Air and Steam

90. **Saturated Air.** The word "saturated," as applied to air, has a different meaning from that of the same word applied to steam. Saturated steam is DRY steam. Saturated air is the wettest kind of air. SUPERSATURATED air is air containing all the moisture it can hold, in the presence of an additional body of moisture.

When air and a sufficiency of moisture are mixed at the temperature t, the mixture contains water vapor, i.e., steam, at the corresponding pressure p_s, which may be taken from the steam table. The total pressure of the moist air is the sum of the partial pressures of dry air and steam. If we are dealing with normal barometer, the partial pressure of dry air is then $p_a = 14.696 - p_s$. The steam is saturated steam and the air is saturated air. One cubic foot of pure dry air unmixed with steam at the temperature t and normal barometer would weigh $(144 \times 14.696) \div (53.36T)$ lb. One cubic foot of air in the present saturated mixture weighs $w_a = 144(14.696 - p_s) \div (53.36T)$ lb., which is less than the weight of dry air unmixed

with steam. The weight of steam intermixed with the 1 cu. ft. of dry air in the mixture is $w_s = 1/v$, where v is the specific volume from the steam table at $t°$ F. By Dalton's law, if we employ partial pressures p_s and p_a as above, 1 cu. ft. of mixture contains both 1 cu. ft. of dry air and 1 cu. ft. of steam.

Prob. 174. What is the weight of a cubic foot of pure dry air at normal atmospheric pressure and 40° F.? (Ans., 0.0794 lb.)

Prob. 175. The pressure of saturated steam at 40° F. is 0.1217 and its specific volume is then 2438. If air at 40° F. and normal barometer is saturated with moisture, state the partial pressures of steam and air. (Ans., 0.1217 and 14.5743.)

Prob. 176. In Prob. 175, state the weights per cu. ft. of air and steam. (Ans., 0.0787 and 0.00041 lb.) What is the density (weight per cu. ft.) of the mixture? (Ans., 0.07911 lb.)

91. **Unsaturated Air.** The "absolute humidity" is the ratio of weight of steam to weight of air. In Prob. 176, it is 0.0052. If there is insufficient moisture present to saturate the air, its temperature will be t, but its pressure will be p_s', LESS than p_s. Hence the moisture in such a mixture will be SUPERHEATED STEAM. Call its weight per cu. ft. w_s'. Then the RELATIVE HUMIDITY is defined as $r_h = p_s'/p_s = w_s'/w_s$. The partial pressure of air in the mixture, at normal barometer, is $14.696 - p_s'$, from which the weight of dry air in a cubic ft. of mixture may be computed as before. The weight of steam will be $r_h w_s$.

Prob. 177. What is the absolute humidity of saturated air at 212° F.? (Ans., ∞.)

Prob. 178. At $t = 200°$, $p_s = 11.52$, $w_s = 0.02976$. For a relative humidity of 0.9, find for moist air the values of p_s' (Ans., 10.368), p_a' (Ans., 4.328), w_s' (Ans., 0.026784), w_a' (Ans., 0.0177).

Prob. 179. As t increases, how will p_s, p_a, w_s and w_a vary? (Ans., p_s and w_s will increase, p_a and w_a will decrease.)

92. **Moist Air Processes.** Following Equation (13), the value of R for a mixture of air and steam is

$$R_m = \frac{R_s w_s' + 53.36 w_a'}{w_s' + w_a'} = \frac{85.8 w_s' + 53.36 w_a'}{w_s' + w_a'},$$

the value of R for steam being taken at 1544/18.016 = 85.8. R_m will often differ notably from 53.36.

When a fixed body of unsaturated moist air is cooled, the value of w_s' does not change, but that of w_s decreases. Hence the relative humidity increases. If the process is carried sufficiently far, the mixture will become saturated air. Beyond this point, the air will be supersaturated and dew will be deposited. Suppose $r_h = 0.60$ at 100° and the mixture is cooled to 90°. The original 1 cu. ft. becomes $(460 + 90)/(460 + 100) = 0.982$ cu. ft. At 90°, the density of saturated steam is 0.002131. The weight of steam in 0.982 cu. ft. of mixture, if saturated, would be $(w_s)_{90} = 0.982 \times 0.002131 = 0.00209$. At 100°, the weight present if saturated is 0.002851. This weight, multiplied by the original relative humidity of 0.60, is still present after cooling to 90°. Hence at 90° the relative humidity will have become

$$r_h = \left(\frac{w_s'}{w_s}\right)_{90} = \frac{0.002851 \times 0.60}{0.00209} = 0.82.$$

Prob. 180. Find the value of R_m in Prob. 176. (Ans., 53.5.)

Prob. 181. In Art. 92, what is the relative humidity if the mixture is further cooled to 85°, where $w_s = 0.001832$? (Ans., 0.96.)

Pure steam at 102° F. has a pressure of 1 lb. absolute. Hence, however low the temperature of condenser circulating water, a low temperature of heat rejection in the steam engine (shown in Art. 36 to be essential for highest potential efficiency) is dependent on the maintenance of a good vacuum by the air pump. By using a mixture of air with steam in an engine, a rejection temperature as low as the circulating water will permit might be attained along with any pressure or vacuum (above the pressure given in the steam table for cooling water temperature) which might be desired. We might have the steam and air at 70° F. The partial pressure of the steam would then be

only 0.3626 lb. per sq. in., with saturated air. The total pressure might however be 1 or 2 lb., a range of values easily maintained by modern vacuum apparatus.

Prob. 182. A mixture of air and steam at 2 lb. pressure has a temperature of 126.15° F. What would be the air pressure if the air were saturated? (Ans., 0.) What are the steam and air pressures if $r_h = 0.5$? (Ans., each 1 lb.) In the latter case, how much is the steam superheated? (Ans., 24.32°.) In a cubic foot of such mixture, state the weights of air and steam. (Ans., 0.00461 and 0.00289 lb.) The mixture is cooled at constant pressure to 70° F. What will then be its volume? (Ans., 0.903 cu. ft.) What weight of steam could be held by this volume of mixture, if saturated? (Ans., 0.001037 lb.) Is it saturated? (Ans., super-saturated.) How much steam must have been condensed as dew? (Ans., 0.001853 lb.) What weights of water, saturated steam and dry air are now present? (Ans., 0.001853 lb. water, 0.001037 lb. dry steam, 0.00461 lb. air.) How much heat above 32° is contained in the water and steam? (Ans., 1.2005 B.t.u.)

CHAPTER V

EFFICIENCY AND POWER OF STEAM ENGINES

Efficiency

93. **Prediction of Efficiency.** Following Art. 82, if an engine uses W_s lb. of steam per hr. while developing M Ihp., its ACTUAL INDICATED THERMAL EFFICIENCY is

$$e_a = \frac{2545}{\frac{W_s}{M}(Q_1 - h_2)} = \frac{2545M}{W_s(Q_1 - h_2)} \tag{54}$$

where Q_1 = total heat above 32° of steam at the throttle condition (wet, dry or superheated) and h_2 = heat of liquid at the pressure existing in the exhaust pipe. Let e denote the efficiency of the Rankine (complete expansion) cycle under the conditions Q_1, h_2. Then the RELATIVE EFFICIENCY, or ratio of actual and ideal efficiencies, is

$$e_R = e_a \div e. \tag{55}$$

The heat rate of the actual engine is

$$\frac{W_s}{60M}(Q_1 - h_2) \text{ B.t.u. per Ihp. min.}$$

The work done by it per lb. of steam is

$$e_a(Q_1 - h_2) = \frac{2545}{W_s \div M} \text{B.t.u.}$$

Prob. 183. Under the conditions of Prob. 158, a certain compound engine uses 13.3 lb. of steam per Ihp.-hr. Find the actual efficiency, relative efficiency, heat rate and work per lb. of steam. (Ans., 0.17: 0.55: 250: 190.)

94. **Probable Values.** A rough estimate of the probable economy of a new engine for definite steam conditions may be formed by using the following values of e_R:

	Simple	Compound	Triple Expansion
Saturated steam, non-condensing	0.6	0.65	—
Saturated steam, condensing	0.4	0.5	0.6
Highly superheated steam	0.65	0.65	—

Best Recorded Performances, Standard Engines

	Saturated Steam, Lb. Per Ihp.-hr.	Steam Superheated Above 150° F., B.t.u. per Ihp.-min.
Simple non-condensing	21	300
Simple condensing	16.3	226
Compound non-condensing	16	245
Compound condensing	12	200
Triple-expansion condensing	10½	190

Records as good as these are rarely attained. The following steam rates may be realized with saturated steam by engines in good condition and operating at their most favorable loads:

	Single Valve	Double Valve*	Four Valve†	Four Valve, Releasing‡
Simple non-condensing	33	30	29	26
Simple condensing	27	23	21½	21½
Compound non-condensing	25½	23	22	22
Compound condensing	20	17	15	15

These rates will be surpassed with very high steam pressures or exceptionally good vacuum.

95. **Indicator Diagram vs. Rankine Cycle.** So far as efficiency is concerned, the Rankine cycle, which represents the operation of the whole power plant, establishes a limit for the engine specifically. A still closer limit is found in the "incomplete expansion" cycle, *ABCDF*, Fig. 28. The actual diagram from the engine appears relatively as *abceg*. The rounding of corners in the actual diagram is due to slow movements of valves. The reduced admission pressure and augmented back pressure are caused by friction of steam flowing through valve ports and

* Separate cut-off control.

† Admission, cut-off, release and compression independently controlled.

‡ Automatic disengagement causing very sharp cut-off.

passages. Clearance exists as a necessary evil in the actual engine. It is relatively large in engines of small size.

The operation of COMPRESSION, *eg*, is chiefly introduced for its cushioning effect, to promote quiet and smooth running. Compression is greatest in high-speed engines. From an economical standpoint its duration should probably never exceed 10 per cent. of the stroke.

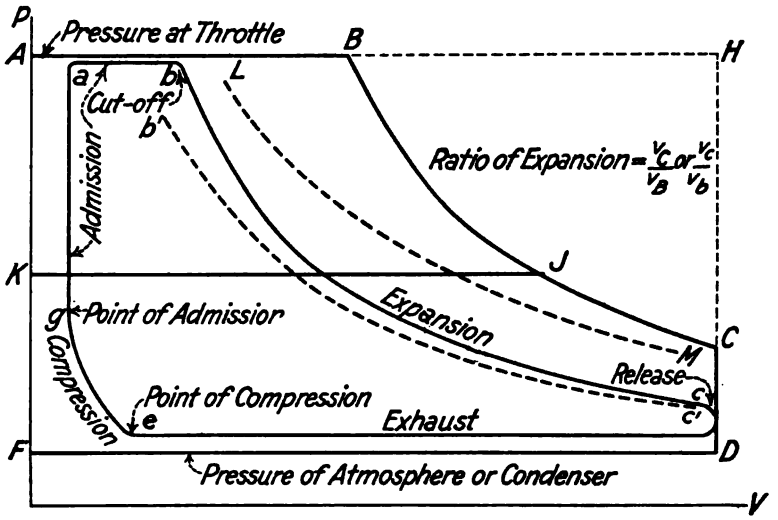

FIG. 28. Action of Steam Engine.

The lines *bc* and *BC* are unlike in slope and position. In particular, the volume at *b* is much less than that at *B*. This is due to CYLINDER CONDENSATION, an important factor in engine economy. If the cylinder were covered with a perfect heat-insulating material, there would nevertheless be a rapid alternation of temperature of its metal walls, according to the temperature of the steam in contact with them. The walls are heated during admission and the early part of expansion and cooled during the late expansion and exhaust. They abstract heat from the incoming steam very rapidly, so that at the point of cut-off the steam is often less than 0.60 dry. This accounts for the

small volume at b. Late in the expansion stroke, the steam has cooled off and the walls of the cylinder now supply it with heat. This raises the right-hand end of the expansion curve so that bc is less "steep" than BC. The restoration never compensates for the initial loss. The availability of heat for doing work depends on its temperature (Art. 36). Low-temperature heat is worth less than the same quantity of high-temperature heat.

The curve bc is therefore not adiabatic. It may be represented by an equation $pV^a = \text{const.}$, a rough average value of a being 1.0. This does NOT make it isothermal, the saturated steam isothermal being a line of CONSTANT PRESSURE. For non-condensing unjacketed engines, $a = (x_b + 0.614) \div 1.258$, nearly.

Condensation is accentuated by high ratios of expansion, which increase the magnitude and duration of the temperature alternations. It is apt to be excessive at slow speeds. It is large in small engines, because their ratio of surface to volume is large. It is reduced in multiple-expansion engines (Art. 100).

Prob. 184. What steam consumption might be expected from the tabular value of e_R in a simple condensing engine under the conditions of Prob. 183? (Ans., 18.1.) How much better than this was the performance of the best standard simple condensing engine? (Ans., 10 per cent. better.) What is the chief reason why an average good engine uses $21\frac{1}{2}$ lb.? (Ans., it uses a lower steam pressure.)

Prob. 185. In Fig. 28, will the rounding of corners be most pronounced at light loads or heavy loads? Will the differences of pressure between ab and AB, ce and DF, be greater at slow speeds or at high speeds? Why should compression be greatest in high-speed engines?

Prob. 186. What is the value of x_b when that of a for the curve bc, Fig. 28, is 1.0? (Ans., 0.644.) What then is the approximate value of $V_b \div V_B$? (Ans., 0.644.) What is a when $x_b = 1.0$? (Ans., 1.28.)

Prob. 187. Assume that condensation varies inversely with speed. The condensation loss is 10 per cent. at 300 r.p.m. What will it then be at 100 r.p.m? (Ans., 30 per cent.)

Prob. 188. The surface of a cylinder is $\pi dL + (\pi/2)d^2 = \pi d(L + (d/2))$ and its volume is $(\pi/4)d^2L$. The ratio of surface to volume is therefore $4/d + 2/L$. If the actual condensation loss is proportional to surface and the weight of steam used is proportional to volume, state a ratio expressing relative condensation loss per lb. of steam used. (Ans., $4/d + 2/L$.) Compare condensation losses in 10 by 20 and 40 by 60 in. cylinders, all other factors being disregarded. (Ans., the condensation loss per lb. of steam in the large cylinder is only 26.7 per cent. of that in the small.)

96. **Ratio of Expansion.** Values used for ratio of expansion in practice are less than those of highest ideal efficiency, because of condensation. We are limited by condensation to comparatively low expansion ratios. Values from 3 to 5 are common in simple engines, and up to 36 in compounds. Too high a ratio leads to large size of cylinders and excessive cost (p_m being low) as well as to excessive condensation. In constant speed engines the ratio of expansion decreases with increased power. The efficiency increases with increase of power up to about the designed load (Fig. 29). The engine should give its best efficiency at the load usually carried. Fig. 29 gives several curves of steam rate against output or ratio of expansion. All show poor economy at high ratios (light loads). Practically all show poor economy at very low ratios (heavy loads). The low points of the curves are the best operating points. A flat curve is particularly desirable where the load is variable.

97. **Jackets.** The STEAM JACKET consists of a hollow space surrounding the cylinder, filled with high-pressure steam. By keeping the cylinder walls hot at all times, it prevents or diminishes condensation. The steam condensed in the jacket is of course chargeable to the engine ($Q_j - h_4$ B.t.u. per lb., where Q_j = total heat above 32° of steam supplied, h_4 = heat of liquid

at the temperature of the jacket drip), but there is usually a net gain. Values of e_R tabulated in Art. 94 may be increased 0.03 to 0.05 if jackets are employed. The jacket is typical of a large class of devices which permit of the saving of fuel by the expenditure of money. It is therefore to be used where fuel

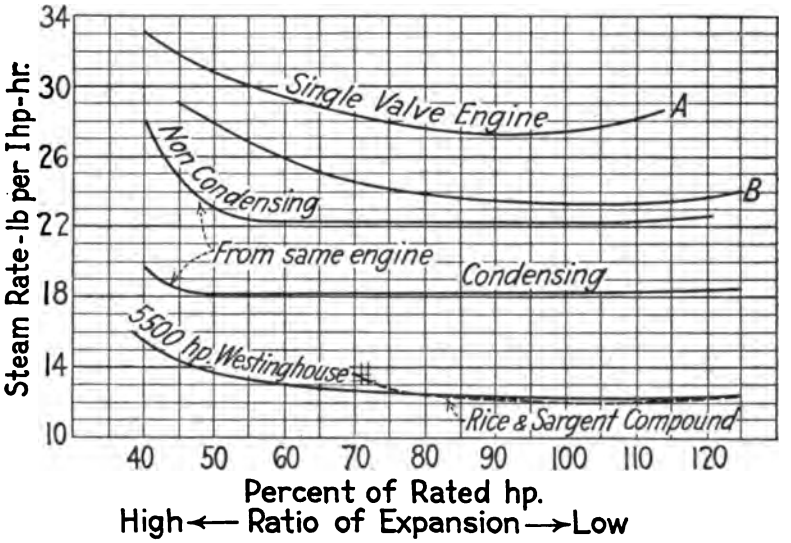

FIG. 29. Variation of Economy with Ratio of Expansion: Constant Speed Engine.

saving is most important: that is, where fuel is costly, or the space which it occupies is valuable, or the load steady. The maximum percentage gain due to jacketing will be realized if the ratio of expansion is high and the speed low. The reason is that these conditions would lead to excessive cylinder condensation if jackets were not used. Compression may be especially small in jacketed engines.

Prob. 189. In Fig. 29, state the per cent. of rated hp. at which best economy is attained, in engines *A* and *B*. (Ans., 90 and 103.)

Prob. 190. The Rice and Sargent engine in Fig. 29 was rated at 700 hp. What weight of steam would it use per hr. when developing 490 hp.? (Ans., 6700 lb.)

Prob. 191. Which engine in Fig. 29 has the lowest steam rate at normal load? Which at 70 per cent. of normal load? At 125 per cent.?

Prob. 192. Under the conditions of Prob. 158, an engine uses 12 lb. of cylinder steam and 1 lb. of jacket steam, both per Ihp.-hr. The jacket supply is dry steam at 240 lb. pressure and its drain temperature is 386°. How many B.t.u. per hr. are chargeable as cylinder steam? (Ans., 13481.2.) As jacket steam? (Ans., 841.7.) Total? (Ans., 14322.9.) What is the actual efficiency? (Ans., 0.178.) The steam rate? (Ans., 13 lb.) The relative efficiency? (Ans., 0.578.)

98. **Superheat.** The thermodynamic advantage of superheat is suggested by Fig. 27. In practice, the steam which expands in the cylinder is usually not superheated, unless superheat exceeding 100° is provided at the boiler. The reason is that the cold cylinder walls cool the steam during admission. Superheat below 100° or 150° simply mitigates condensation. Higher superheat gives the additional advantage of a more efficient cycle. In Fig. 28, if *ABCDF* is the ideal cycle and *abceg* the actual cycle realized with saturated steam, superheat (even slight superheat) moves the point *b* toward *L* and gives an expansion curve like *LM*, "flatter" than the curve *bc*. Since with superheat we should use a larger ratio of expansion, the actual curve, representing the behavior of reduced weight of steam, lies like *b'c'*. The mean effective pressure is then less when superheat is used. Fig. 30 shows the effect of low super-

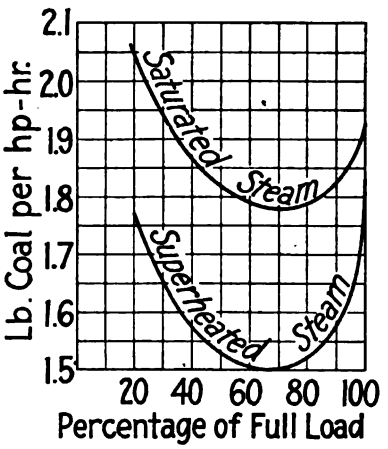

FIG. 30. Effect of Low Superheat (72° to 92°) on Overall Economy. H. M. S. Brittania.

heat. Fig. 31 gives average results at low and high superheat from tests of 40 famous engines. Superheat exceeding 200° may be expected to produce a saving of 20 per cent. in simple engines and 16 per cent. in compounds. The simple engine with superheated steam is often as economical as the compound with saturated steam. The conditions which make superheat desirable are the same as those which favor the use of jackets (Art. 97). The two should not be used together. Low compressions should be used when superheat is employed.

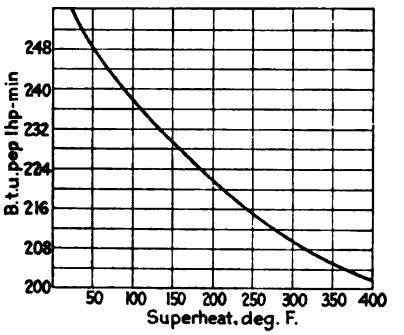

FIG. 31. Relation Between Efficiency and Superheat. Compound Condensing Engine.

Prob. 193. From Fig. 31, what is the gain by using 400° superheat as compared with 100°? (Ans., 15 per cent.)

Prob. 194. In Fig. 30, what was the load of best economy? (Ans., 70 per cent. of full load.) With saturated steam, how much more total coal is used per hr. at full power than at best economy? (Ans., 54 per cent.) Note that in warships engines are designed for best economy at reduced power and speed.

99. **Speed.** If an engine duplicated its own indicator diagram continuously, the mean effective pressure would be constant, and the ihp. would be directly proportional to the r.p.m. In single-valve engines with fixed cut-off, the valve action at high speeds is inferior. Hence the amount of steam used tends to increase somewhat more rapidly than the ihp. This is just about offset by the favorable influence of high speed on cylinder condensation. The higher the speed, the less time is available for transfer of heat. From this standpoint, the steam consumption should increase a little LESS rapidly than the Ihp., as the speed increases. As a final result, when the total hourly steam

consumption is plotted against Ihp., the resulting curve is (for engines of this type) a straight line. The steam RATE (lb. per Ihp.-hr.) is therefore best (lowest) at high speeds.

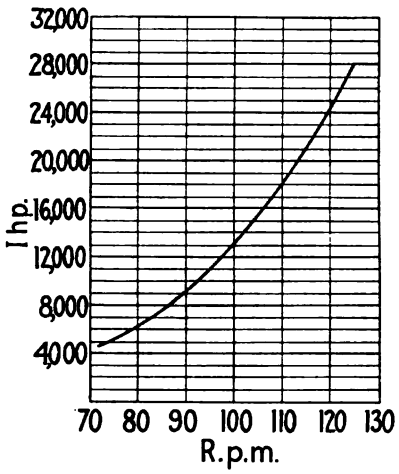

FIG. 32. Indicated Power of an Armored Cruiser at Various Speeds.

In marine engines, with reversing valve gears, cut-off is delayed at high speed, which is the full-power condition. The power therefore increases faster than the speed (Fig. 32). The most economical point of cut-off, and of the ratio of expansion, are fixed at maximum speed for merchant vessels and at reduced cruising speed for warships. For the former, the steam rate per Ihp. hr. should decrease as the speed increases. For the latter, it should vary somewhat like the coal rate of Fig. 30.

Prob. 195. An auxiliary engine uses 1730 lb. of steam per hr. when developing 100 hp. and 11 730 lb. when developing 500 hp. The curve of hourly steam against hp. is a straight line. What is the steam rate when developing 300 hp.? (Ans., 22.43 lb. per hp.-hr.)

Prob. 196. In Fig. 32, compare mean effective pressures at 90 and 120 r.p.m. (Ans., p_m at the low speed is 51 per cent. of that at the high speed.)

100. **Multiple Expansion.** By partially expanding the steam in one cylinder and then conducting it to another, the alternations of temperature in any one cylinder are reduced. Hence condensation is diminished,* and a much larger total ratio of ex-

* The temperature range in any cylinder is halved, but the total size (surface) of cylinders is not doubled.

pansion may be utilized with advantage. These ratios range up to 30 or more in compound (two-stage) and triple expansion engines. They are rarely over 5 in simple engines. There is a corresponding gain in economy. The compound will have a steam rate 20 to 30 per cent. lower than the simple engine. The triple may be slightly better still. Four stages (quadruple expansion) are scarcely to be considered for naval service. The number of cylinders is often greater than the number of expansive stages, in order to avoid the use of very large low pressure cylinders. Compound engines usually have small compression. They should always have high initial pressure and should run condensing. The REHEATER is a vessel placed between the cylinders. Steam passes through it on the way from one cylinder to the next. It is equipped with coils containing high-pressure steam. This dries and often superheats the steam which is on its way to the second stage. The heat supplied is computed in the same way as the heatsupplied at a jacket. The argument in favor of the use of a reheater is the same as that for the use of superheat in any cylinder.

Prob. 197. From the last table in Art. 94, state the percentage gain by compounding in four valve engines, (*a*) condensing, (*b*) non-condensing. (Ans., (*a*), 30: (*b*) 24.)

101. **Quality of Exhaust Steam.** The heat supplied to an engine, per lb. of steam, is $Q_1 - h_2$: Art. 93. The work done is

$$\frac{2545}{W_s \div M} = \frac{2545 \times \text{Ihp.}}{W_s} \text{ B.t.u. per lb. of steam.}$$

The heat REJECTED is the difference of these two quantities. It is equal (ignoring radiation) to $(xL)_E$, where the subscript $_E$ refers to the condition of steam in the exhaust pipe. Then

$$(xL)_E = Q_1 - h_2 - \frac{2545}{W_s \div M}. \tag{56}$$

These relations are important in condenser design. The condenser may however absorb more heat than that indicated by Equation (56), because it may cool the condensation.

Prob. 198. In Prob. 183, how much of the heat in each lb. of steam is rejected to the condenser? What is the dryness of the steam in the exhaust pipe? (Ans., 933.44 B.t.u.: 0.90.) If the condenser cools the condensed steam to 90°, how much additional heat does it remove? (Ans., 11.8 B.t.u. per lb.)

102. **Desirable Back Pressure.** Even from a purely thermodynamic standpoint there is a limit to the condenser vacuum desirable in simple engines. In Fig. 33, *fbcde* is an incomplete expansion cycle. Consider the effect of lowering the back pressure, so that the cycle becomes *abcdg*. The gain of work is *afeg*. The increased cost of heat is $h_f - h_a$. The value of *afeg* is from the pV diagram, practically, $(v_e/5.403)$ $(p_f - p_a)$ B.t.u. Now for engines of any importance, v_c is never greater than 4. This volume is the volume of dry (or nearly dry) steam at the throttle pressure. In simple engines, the ratio of expansion is practically never over 5. Now v_e is equal to v_c multiplied by the ratio of expansion. Hence a maximum value of v_e is 20, and a maximum value of *afeg* is $3.7(p_f - p_a)$, in simple engines. If $p_f = 2$, $p_a = 1$, *afeg* = 3.7 B.t.u. But, then, $h_f = 94$, $h_a = 69.8$, heat cost = 94 − 69.8 = 24.2 B.t.u. The efficiency at which we have gained the additional area *afeg* is then 3.7 ÷ 24.2 = 0.152. This may easily be less than the efficiency of the original cycle *fbcde*: hence the lower back pressure (improved vacuum) will LOWER the efficiency of the whole cycle. Simple engines should operate at 2 lb. back pressure (26 in. vacuum). With compound engines, the ratio of expansion is much greater and a better vacuum is desirable.

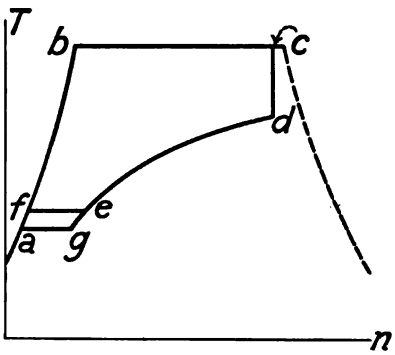

FIG. 33. Desirable Back Pressure.

Prob. 199. A compound engine uses dry steam at 250 lb. pressure and a ratio of expansion of 30. Find v_e. (Ans., 55.5.)

How much work is gained per lb. of steam by lowering the back pressure from 2 lb. to 1 lb.? (Ans., 10.3 B.t.u.) At what efficiency is this quantity of work derived? (Ans., 0.42.)

103. **Results by Special Engines.** The UNIFLOW engine has unidirectional flow of steam, which is supposed to reduce condensation. There is no exhaust valve, the long piston uncovering slots in the cylinder wall when near the end of its stroke. Fig. 34

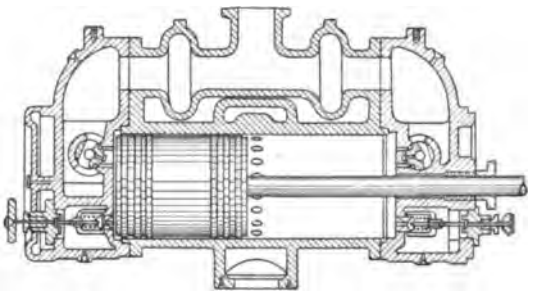

FIG. 34. Uniflow Engine.

shows an engine with Corliss valves. Poppet valves are commonly used. The engine is not well adapted for running noncondensing on account of the excessive compression which then results. A simple unjacketed engine of this type has given the remarkable steam rate of 9.06 lb. with 155 lb. pressure and 667° steam temperature (implying considerable superheat).

The BINARY VAPOR ENGINE uses two fluids. Steam is used in an ordinary cylinder. It then passes to a condenser, in which the cooling fluid is not water, but a volatile liquid (ether or SO_2). This "binary fluid," evaporated in the steam condenser, is used in a second cylinder and then discharged to a surface condenser employing water as a cooling agent. The maximum efficiency is determined by the cooling water temperature and is ideally somewhat higher than that attainable by steam alone with present vacuum apparatus. Actual tests have given a heat rate around 167 B.t.u. per Ihp.-min.

REGENERATIVE ENGINES have occasionally been built. Fig. 35 illustrates the action in four stages (quadruple expansion).

The path *cd* represents adiabatic expansion in the first cylinder. The steam then passes to a reheater (Art. 100) on its way to the second cylinder. A portion of the steam is drawn off from

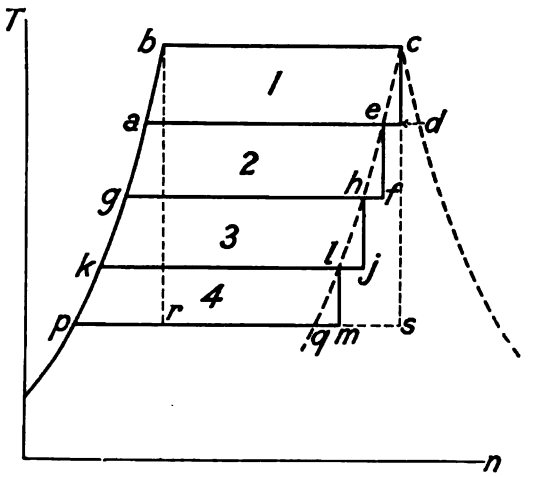

FIG. 35. Quadruple Expansion Regenerative Engine.

this reheater and employed to heat boiler feed water. We may regard the heat withdrawn as the area under *de*, and the heat transferred to the feed water as the (ideally equal) area under *ga*. Similarly after expansion in the second cylinder, the heat under *hf* becomes the heat under *kg*: and after a third expansion the heat under *lj* becomes the heat under *pk*. With an infinite number of stages, the equivalent expansion would be *cq*, the heat under which would equal the heat under *pb*. This is a REGENERATIVE CYCLE, like that in Art. 62, and the only heat chargeable is that under *bc*. The efficiency is ideally $(T_b - T_p)/T_b$, that of the Carnot cycle. Actual tests gave heat rates as low as 169 B.t.u. per Ihp.-min.

Prob. 200. A Uniflow engine has 2 per cent. clearance and opens its exhaust ports at 95 per cent. of the stroke. If the compression curve is pv = const., find the pressure at the end of the compression, the exhaust pressure being 1 lb. What is it if the exhaust pressure is 15 lb.? (Ans., 48½ lb., 727½ lb.)

Prob. 201. A Uniflow engine uses 9.5 lb. of steam per Ihp.-hr. at 150 lb. pressure, superheated 300°. The back pressure is 0.505 lb. Find the heat rate. (Ans., 206 B.t.u. per Ihp.-min.) Is any other value as low as this for a simple engine given in this chapter?

Prob. 202. What is the ideal CARNOT efficiency for steam at 200 lb. initial pressure (dry) at 1 lb. back pressure? (Ans., 0.332.) If by using a binary vapor the rejection temperature is reduced to 75° F., what is then the Carnot efficiency? (Ans., 0.363.) If the relative efficiency AS COMPARED WITH THE CARNOT CYCLE is 0.45, what is then the heat rate? (Ans., 258 B.t.u. per Ihp.-min.)

Prob. 203. Prove, in Fig. 35, that $pbr = qcs$. Prove that the efficiency is that of the Carnot cycle.

Power or Capacity

104. **Mean Effective Pressure, Simple Engine, Saturated Steam.** Methods of finding p_m for ideal cycles have been given in Arts. 80, 81, 84, 85, 87. The USUAL method is based on the diagram $ABCDF$, Fig. 28, the curve BC being assumed to be, $p_B V_B = p_C V_C$. Then

$$p_m = \frac{\Sigma W(\text{ft. lb.})}{144 V_C} = \left(p_B V_B + p_B V_B \log_e \frac{V_C}{V_B} - p_D V_D \right) \div V_C$$

$$= p_B \frac{V_B}{V_C} \left(1 + \log_e \frac{V_C}{V_B} \right) - p_D, \qquad (57)$$

in which p_B = initial pressure, p_D = back pressure, V_C/V_B = ratio of expansion. The maximum overload capacity (with no cut-off) is

$$\frac{AHDF - ABCDF}{ABCDF} = \frac{P_B - P_D - p_m}{p_m}. \qquad (58)$$

Values of p_m from Equation (57) are multiplied by a DIAGRAM FACTOR, f, (sometimes called the "card factor") to give the probable mean effective pressure in the actual engine. Note that $f = abceg \div ABCDF$. Values of f range from 0.6 to 0.9, de-

pending chiefly on the valve gear. Jackets increase f by 0.05 to 0.15. Its value is decreased by high ratios of expansion, speeds above 225 r.p.m., excessive clearance, and inadequate valve passages.

Piston speeds are usually between 500 and 1000 ft. per min. The allowable r.p.m. depend chiefly on the valve gear. Long strokes favor low clearances. See Art. 7.

Prob. 204. With an initial pressure of 115 lb., a back pressure of 2 lb. and a ratio of expansion of 4.5, find p_m and ideal overload capacity. If $f = 0.85$, piston speed = 800, r.p.m. = 100, find the cylinder diameter and stroke, the engine being rated at 500 Ihp. (Ans., $p_m = 62$, ideal overload capacity = 82 per cent., diameter = 22.3 in., stroke = 48 in.)

105. **Superheated Steam.** For superheated steam, the expansion curve BC, Fig. 28, is "steeper" than for saturated steam. Its equation is $p_B V_B{}^a = p_C V_C{}^a$, where a may have values about as follows:

Ratio of Expansion	Superheat, °F.						
	200	250	300	350	400	450	500
	Values of Exponent a						
8.0	1.05	1.07	1.09	1.11	1.13	1.15	1.18
7.0	1.06	1.08	1.10	1.12	1.14	1.16	1.19
6.0	1.07	1.09	1.11	1.14	1.16	1.18	1.21
5.0	1.08	1.10	1.12	1.15	1.17	1.19	1.22
4.0	1.09	1.11	1.13	1.16	1.18	1.20	1.23
3.0	1.10	1.12	1.15	1.17	1.20	1.22	1.25

The mean effective pressure is now

$$p_m = p_B \frac{V_B}{V_C} \cdot \frac{a}{a-1} - p_B \left(\frac{V_B}{V_C}\right)^a \cdot \frac{1}{a-1} - p_D. \qquad (59)$$

This gives a lower value than Equation (57)—see Art. 98—but (as with jackets) the value of f is higher. With SLIGHT superheat, use Equation (57) for p_m but use high values of f in connection therewith.

Prob. 205. Under the conditions of Prob. 204, what is the value of p_m for steam superheated 100°? (Ans., 62.) If the steam is superheated 300°, what is the probable value of a? Of p_m? (Ans., $a = 1.125$, $p_m = 59$.)

106. **Compound Engine: Ratios.** In the preliminary proportioning of a compound engine, it is assumed that the combined diagrams of the two stages will appear as $ABCDF$, Fig. 28, and that the steam exhausted from the high-pressure cylinder along JK enters the low pressure cylinders along KJ without fall of pressure or change of volume. The curve BC is assumed to be (with saturated steam) $p_B V_B = p_C V_C$. The ratio of cylinder volumes (CYLINDER RATIO) is $C = V_C/V_J = p_J/p_C$. For cylinders of equal strokes (the usual case), $C = A_l/A_h$, where A_l and A_h are the cross-sectional areas of low and high pressure cylinders. But $p_C = p_B/r$, where r = whole ratio of expansion. Then $C = rp_J/p_B$. The RECEIVER PRESSURE (receiver between the cylinders) is p_J. The ratio of expansion in the high pressure cylinder is $r_h = V_J/V_B = p_B/p_J$. That in the low-pressure cylinder is $V_C/V_J = C$.

There are five methods of establishing a value for p_J.

(1) It may be assumed.

(2) A value may be assumed for C. Then $p_J = Cp_B/r$.

(3) The temperature ranges may be equalized in the two stages. Then $t_B - t_J = t_J - t_D$, $t_J = (t_B + t_D)/2$. Values of t_B and t_D are those from the steam table, corresponding with the pressures p_B and p_D. The pressure p_J is then the tabular pressure for t_J.

(4) Maximum total piston pressures may be equalized, i.e.,

$$\begin{aligned} p_B - p_J &= C(p_J - p_D) \\ p_J &= p_B - C(p_J - p_D) \\ &= p_B - \frac{rp_J}{p_B}(p_J - p_D) \\ &= p_B{}^2 \div (p_B + rp_J - rp_D). \end{aligned} \tag{60}$$

THIS IS THE IMPORTANT METHOD FOR MARINE PRACTICE.

(5) The two stages may develop equal power. Then $ABJK = KJCDF$, or

$$p_B V_B \log_e \frac{p_B}{p_J} = p_J V_J + p_J V_J \log_e \frac{V_C}{V_J} - p_D V_D$$

$$\log_e p_B - \log_e p_J = 1 + \log_e C - r\frac{p_D}{p_B}$$

$$= 1 + \log_e p_J - \log_e p_B + \log_e r - r\frac{p_D}{p_B}$$

$$\log_e p_J = \frac{1}{2}\left(\log_e r + \frac{p_D}{p_B} r - 1\right) - \log_e \frac{r}{p_B} \qquad (61)$$

Prob. 206. Given initial and back pressures of 120 and 2 lb., with $r = 25$, find p_J under each of the following conditions:

(*a*) $C = 4$. (Ans., 19.2.)

(*b*) Temperature ranges equalized. (Ans., 22.)

(*c*) Maximum total pressures equalized. (Ans., 22.7.)

(*d*) Power division equalized. (Ans., 17.8.)

107. **Approximate Dimensions.** After having determined p_J by one of the foregoing methods, the mean effective pressures in the high and low pressure cylinders are

$$\left.\begin{aligned} p_{mh} &= \frac{ABJK}{V_J} = \frac{p_B}{r_h}\log_e r_h \\ p_{ml} &= \frac{KJCDF}{V_D} = \frac{p_J}{C}(1 + \log_e C) - p_D \end{aligned}\right\}. \qquad (62)$$

The output of the engine is

$$\text{hp.} = \frac{fS}{33\,000}(p_{mh}A_h + p_{ml}A_l) = \frac{fSA_l}{33\,000}\left(\frac{p_{mh}}{C} + p_{ml}\right), \qquad (63)$$

from which A_l is determined. The value of f is about the same as that in a simple engine using a ratio of expansion equal to $\sqrt{r}$. The quantity $(p_{mh}/C + p_{ml})$ is called the TOTAL MEAN EFFECTIVE PRESSURE REFERRED TO THE LOW-PRESSURE CYLINDER. Its value is the same as that given by Equation (57), applied to the whole diagram $ABCDF$. Then

$$hp = \frac{fSAl}{33\,000}\left\{\frac{p_B}{r}(1 + \log_e r) - p_D\right\}. \qquad (63a)$$

It thus appears that THE SIZE OF THE HIGH PRESSURE CYLINDER IS WITHOUT INFLUENCE ON THE TOTAL POWER OF THE ENGINE.

Prob. 207. The initial, back and receiver pressures are 120,2 and 17.8. The ratio of expansion is 25. The engine develops 500 Ihp. at 800 ft. piston speed and 100 r.p.m. Find cylinder diameters and strokes. Use $f = 0.9$. (Ans., 20.8 and 40 in. by 48 in.)

Prob. 208. In Prob. 207, what are the mean effective pressures in each of the two cylinders? (Ans., 33.9 and 9.11.) Show that the two cylinders DO develop equal power.

Prob. 209. If in the foregoing TWO low pressure cylinders are used, state dimensions. (Ans., all strokes 48 in.: diameters, high pressure, 20.8 in.; two low pressure, 28.5 in.)

Prob. 210. Given $p_{mh} = 35$, $p_{ml} = 9$, $C = 4$, in a compound engine. What proportion of the total power is developed in each cylinder? (Ans., high pressure 49.2 per cent., low pressure 50.8 per cent.)

108. **Cylinder Ratio and Drop.** The value of C has not been thoroughly standardized. It should increase with r. High values increase the initial steam pressure in the second stage. The margin of overload capacity, obtainable by admitting reduced pressure steam from the boilers to the low-pressure cylinder, is thus decreased. Values up to 6 or 7 have given highly efficient engines. The ideal diagram of Fig. 28 shows terminal DROP, CD, in the low pressure cylinder but none in the high pressure cylinder. DROP (high pressure drop) is often provided for. This reduces the value of f about 0.05. In general practice, the mean effective pressure referred to the low pressure cylinder ($p_{mh}/C + p_{ml}$) is kept around $1.2 + 0.12p_B$.

Prob. 211. Following the rule in Art. 108, what would have been the diameter of the low pressure cylinder for 500 Ihp. at 800 ft. piston speed with $f = 0.9$? (Ans., 43 in.)

109. **Compounds with Superheat.** For superheat below 150°, the foregoing method may be used, increasing f as when jackets are employed. With high superheat, take the value of a in Art. 105 which would be suitable for a ratio of expansion equal to $\sqrt{r}$ in a simple engine. Then, following Equation (59)

$$\left.\begin{aligned} p_{ml} &= \frac{p_J a}{C(a-1)} - \frac{p_J}{C^a(a-1)} - p_D \\ &= \frac{p_J}{a-1}\left(\frac{a}{C} - \frac{1}{C^a}\right) - p_D \\ p_{mh} &= \frac{p_B}{a-1}\left(\frac{a}{r_h} - \frac{1}{r_h{}^a}\right) - p_J \end{aligned}\right\} \tag{64}$$

In this case $p_J = p_B(C/r)^a$ and $r_h = r/C$. If the work of the two cylinders is to be equally divided, it will be found that

$$C = \left[\frac{1}{2a}\left\{\frac{p_D}{p_B}r^a(a-1) + ar^{a-1} + 1\right\}\right]^{1/(a-1)}.$$

The horse power is

$$\text{hp.} = \frac{fSA_l}{33\,000}\left(\frac{p_{mh}}{C} + p_{ml}\right),$$

where the quantity in the parenthesis has the value given by Equation (59).

Prob. 212. For $r = 25$ at 300° superheat, what value of a should be used? (Ans., 1.12.) If $p_B = 120$, $p_D = 2$, $p_J = 20$, find C. (Ans., 5.05.) Find the value of C for equality of work. (Ans., 5.2.) What is then the value of p_J? (Ans., 20.6.) Under the latter condition, find r_h, p_{mh}, p_{ml}. (Ans., $r_h = 4.8$, $p_{mh} = 40\frac{1}{2}$, $p_{ml} = 7.85$.) What is the value of the total mean effective pressure referred to the low pressure cylinder, as determined by Equation (59)? (Ans., 15.7.)

110. **Triple Engines: General Case of Multiple Expansion.** For any multiple expansion engine using saturated steam (or steam superheated less than about 150° F.), the size of the low pressure cylinder is given by Equation (63a). The sizes of the other cylinders, for zero drop, are then determined from assumed

values of cylinder ratios or receiver pressures. Their sizes influence efficiency but not power.

With high superheat, similarly, following Equation (59),

$$\text{hp.} = \frac{fSA_l}{33\,000}\left(\frac{p_B}{r}\cdot\frac{a}{a-1} - \frac{p_B}{r^a(a-1)} - p_D\right). \qquad (65)$$

If b is the number of expansion stages, take a from Art. 105 for a ratio of expansion of $\sqrt[b]{r}$.

Prob. 213. The battleship Oklahoma uses triple engines at 280 lb. initial absolute pressure. Assuming 1 lb. back pressure, and 10 000 Ihp. capacity at cruising speed when $r = 30$, with $f = 0.9$, what would be the diameter of each of the two low pressure cylinders at 1000 ft. piston speed, using saturated steam? (Ans., 76 in.)

Prob. 214. The cylinder ratios in the Oklahoma are 1 : 2.84 : 9.93 (high — intermediate — low). Find the diameters of high and intermediate cylinders under the conditions of Prob. 213. (Ans., 34 and 64 in.)

Prob. 215. Under the conditions of Prob. 213, but with steam superheated 300°, what is the probable value of a? (Ans., 1.15.) What is then the diameter of the low pressure cylinder? (Ans., 84½ in.)

CHAPTER VI

VAPOR REFRIGERATION

111. **Ideal Cycle.** The ideal cycle in vapor refrigeration is a Rankine cycle worked counter-clockwise (Fig. 36). Assume the fluid to be at low temperature and pressure, somewhere on the line *eb*, but nearer *e* than *b*, i.e., in the condition of a very wet vapor. It is contained in coils in the room or tank to be cooled, the contents of such room or tank being warmer than the fluid. Heat will flow from such contents to the fluid. This will not increase the temperature of the fluid, but will vaporize whatever liquid it contains.

When the vaporization is nearly (but not quite) complete, allow the fluid to flow out to a COMPRESSOR. Here it is compressed adiabatically until its temperature is above that of an available supply of cooling water. The operation is indicated by the path *bc*. The pressure will rise, as well as the temperature. The fluid now passes to the CONDENSER, where it is subjected to the cooling effect of water in the coils. This water condenses the vapor (path *cd*) but in general cannot cool it much. The final operation necessary is a reduction in temperature of the resulting liquid. In air refrigerating machines (Arts. 57, 60) the final operation took place in an EXPANDING CYLINDER With air, mere expansion from a high pressure to a lower pressure would not produce cooling, unless at the same time work is done. With a vapor, on the other hand, expansion alone will reduce the temperature. Hence the final operation, which in Fig. 36 reduces the temperature of the liquid from t_d to t_e, is merely a reduction of pressure accomplished by allowing the fluid to flow through an EXPANSION VALVE. A high pressure exists from the compressor outlet to the expansion valve inlet. A low pressure exists from the expansion valve outlet through the cold room to the compressor inlet.

112. **Choice of Fluid.** It is desirable that the HIGH pressure (necessarily the pressure corresponding with a normal cooling

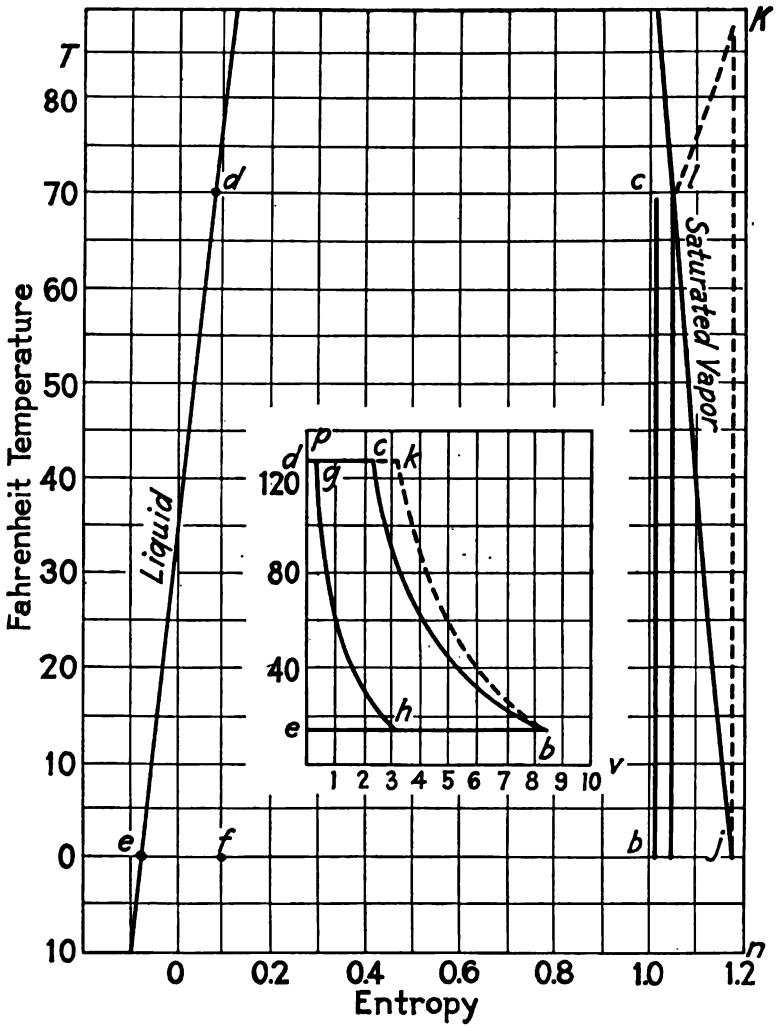

FIG. 36. Vapor Refrigeration with Ammonia.

water temperature of say 80° F.) be not TOO high. It is also desirable that the low pressure (that corresponding with the

lowest temperature of the fluid, which for ice-making and general service is about 0° F.), be not TOO low. The latter pressure should preferably be ABOVE the pressure of the atmosphere. If it is lower than this, any leakage will be inward. Leakage cannot then be readily detected, and the fluid will be diluted if leakage occurs. Fig. 20 shows that from the first standpoint the best fluids are ether, ethyl chloride, sulphur dioxide and ammonia. Carbon dioxide, at 80° F., gives a pressure of nearly 1000 lb. per sq. in. From the second standpoint, ether, ethyl chloride and SO_2 are all unsatisfactory, giving pressures below that of the atmosphere at 0° F. The fluids mostly used are SO_2, ammonia and CO_2. The last is most popular in marine service, since it is innoxious. In the following discussion, reference is made particularly to ammonia. This is for two reasons. Its properties are fairly well known. It permits greater flexibility of cycle, because there is in no case any approach to its critical temperature (Art. 65).

113. **Action in the Expansion Valve.** A new type of vapor process is here encountered, which will be more fully discussed later (Chapter VIII). The liquid at *d*, Fig. 36, is converted to fluid at the pressure p_e WITHOUT CHANGE OF HEAT CONTENT. Since $h_d > h_e$, the fluid cannot be liquid at the new pressure. It can only be a wet vapor. Its state is *f*, defined by HEAT AT f = HEAT AT *d*, or $h_f + (xL)_f = h_d$. We must regard the process as being represented by *def*. It is NOT an adiabatic process. The refrigeration that can be performed by the fluid is then NOT the area under *eb*, but ONLY THE AREA UNDER *fb*.

Properties of Saturated Ammonia Vapor

t Temperature, Deg. F.	p Pressure, Lb. per Sq. In.	v Volume, Cu. Ft. per Lb.	h	L	H	r	s_w	s_f	s_g
0	29.95	9.190	−33.7	572.2	538.5	521.4	−0.0709	1.2450	1.1741
70	129.20	2.296	42.1	512.8	555.0	458.5	0.0813	0.9684	1.0497
90	181.80	1.650	65.3	493.5	558.9	438.9	0.1238	0.8981	1.0219
95½	199.00	1.512	71.9	488.0	559.9	433.3	0.1356	0.8790	1.0146

(From Goodenough's "Properties of Steam and Ammonia")

Prob. 216. Liquid ammonia at 70° expands to 0°. What is the dryness of the resulting vapor? (Ans., 0.132.) What is its entropy? (Ans., 0.0936.)

Prob. 217. This vapor in Prob. 216 is 0.87 dry when it leaves the coal room. How much refrigeration has been done per lb. of fluid? (Ans., 422 B.t.u.)

114. **Ideal Power and Efficiency.** In symbols, per lb. of fluid, and in B.t.u.,

Refrigeration done = heat under $fb = L_{fb}(x_b - x_f)$

Heat rejected at condenser = heat under $cd = L_{cd}(x_c)$

Work expended = heat at c − heat at $b = (h + xL)_c - (h + xL)_b$

$$\text{Efficiency} = \frac{\text{Refrigeration}}{\text{Work}} = \frac{L_{fb}(x_b - x_f)}{(h + xL)_c - (h + xL)_b}. \quad (66)$$

The last may exceed unity. The value of x_c is found from

$$(n_w + xn_e)_c = (n_w + xn_e)_b.$$

Efficiencies are much higher than with "dense air" machines (Art. 60). The mean effective pressure is

$$p_m = \frac{\text{work} \times 5.403}{\text{maximum volume}}: \text{ see Equation (67).}$$

Prob. 218. The fluid is ammonia. Temperature limits are 0° and 70°. The fluid is 0.87 dry at the beginning of compression. Find the dryness at the end of compression. (Ans., 0.96.)

Prob. 219. In Prob. 218, how much heat is removed by the circulating water at the condenser? (Ans., 494 B.t.u. per lb. of fluid.)

Prob. 220. In Prob. 218, how much work is expended? (Ans., 70.8 B.t.u. per lb. of fluid.) What is the ideal efficiency? (Ans., 5.98.)

115. **Capacity, Power and Size.** If T = tonnage (Art. 59), the necessary refrigeration is $200T$ B.t.u. per min. Then the weight of fluid to be circulated per minute is $w = 200T \div$ REFRIGERATION PER LB. and the ideal hp. is $(w/42.42)\{(h + xL)_c$

$-(h + xL)_b\}$. The heat rejected at the condenser is $wL_{cd}x_c$ B.t.u. per min. A close approximation to the compressor displacement may be made as follows:

The volume of fluid present at the beginning of compression is v_b in the pv diagram. (Lettering on the pv diagram corresponds with that on the Tn diagram.) This is substantially $(xv)_b$. Assume clearance $= c$ and assume that the clearance expansion curve is $p_g v_g = p_h v_h$. Then $v_h = cD(p_g/p_h)$, where D = displacement of compressor. The volume of fresh fluid drawn in and compressed is $v_b - v_h = (1 + c)D - cD(p_g/p_h)$ $= D\{1 + c(1 - (p_g/p_h))\}$. The weight, w, is this quantity divided by $(xv)_b$, the specific volume of the wet vapor. The weight being known, the displacement may be found.

Thus, suppose clearance to be 4 per cent., and $p_g/p_h = 5$. The volume compressed is $D\{1 + 0.04(1 - 5)\} = 0.84D$. Note that 0.84 is the VOLUMETRIC EFFICIENCY (Art. 42), suction friction being ignored. Suppose the vapor to be 0.90 dry at the beginning of compression, when its temperature is 0°. Then $(xv)_b = 0.90 \times 9.19 = 8.271$. Suppose 100 lb. of fluid to be circulated per min. Then the volume to be circulated is 827.1 cu. ft. $= 0.84D$ whence $D = 983$ cu. ft. per min.

The equation given for mean effective pressure in Art. 114 may now be written

$$p_m = \frac{5.403\{(h + xL)_c - (h + xL)_b\}}{(xv)_b} \quad (67)$$

Prob. 221. In Probs. 218–220, the capacity of the machine is 100 tons. Find the weight of fluid per min. and the ideal hp. (Ans., 47.4 lb., 79 hp.)

Prob. 222. In the foregoing, how much heat must be carried away by the cooling water? (Ans., 23 400 B.t.u. per min.)

Prob. 223. In Prob. 221, the clearance is 5 per cent., the piston speed 240 ft. per min., the compressor double-acting at 60 r.p.m. Find the diameter and stroke of the compressor cylinder. (Ans., 18.6 and 24 in.) What is the mean effective pressure? (Ans., 48 lb. per sq. in.)

Prob. 224. In Prob. 221, what is the ideal ice-melting effect per Ihp.-hr.? See Art. 59. (Ans., 106.)

Actual Performance. In the foregoing illustrative case, the condenser temperature (70°) is lower than may in general be safely assumed. Consequently the ideal efficiency is higher than is to be expected in ordinary plants. A narrow temperature range leads to efficiency in refrigeration. An ideal ice-melting effect (Art. 59) of about 80 lb. per Ihp.-hr. is attainable between limits of 0° and 90°. About 60 lb. would be considered a fairly good result if actually attained in ordinary practice, with these limits. There is thus implied a relative efficiency of 60/80 = 0.75. This is higher than the value of relative efficiency attainable with steam engines, mainly because the temperatures involved are closer to those of the surrounding atmosphere. The horse power of the engine which drives the compressor is greater than the horse power here computed, the ratio of the two being the MECHANICAL EFFICIENCY. Under the 0°–90° limits, one horse power in the ammonia compressor cylinder should give a trifle less than one ton of capacity. Under unfavorable conditions, as much as 2 hp. per ton may be required at the steam engine cylinder. (Capacity is ice-melting effect, NOT ice-making capacity.)

Piston speeds range from 125 to 600 ft. per min., being low in small machines. Mean effective pressures vary widely (30 to 90 lb.) with the assigned temperature limits.

In purchasing a compressor, the stated tonnage should be checked against the displacement, for the temperature limits to be employed. It is seldom safe to assume a condenser temperature below 90° for year-round shore service, but 75° may be realized on ships in northern latitudes. A low temperature of 0° will just about answer for ice-making with DIRECT EXPANSION (ammonia circulated in coils in the freezing tank). A slightly lower temperature still will accelerate the freezing. When the BRINE SYSTEM is used (ammonia cools the brine and brine does the refrigeration) the possibly offensive fluid is kept away from substances it might injure, but efficiency is sacrificed. The low-

est ammonia temperature must be below the lowest brine temperature, and the latter must be below the desired cold-room

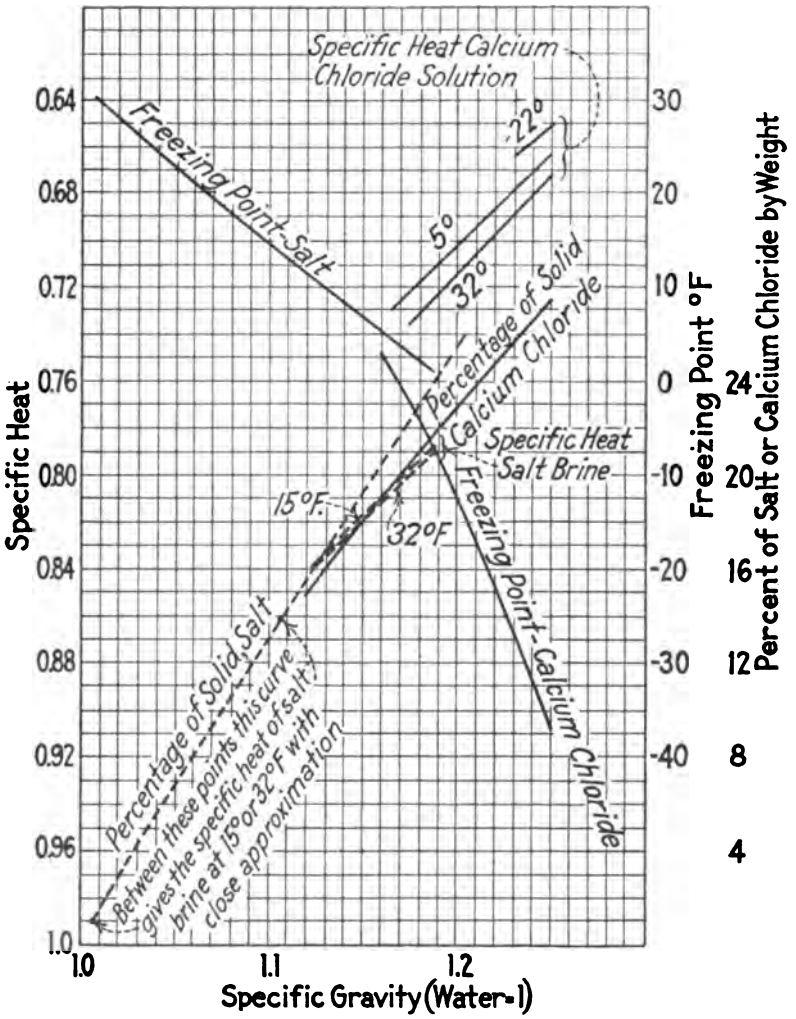

FIG. 37. Properties of Brines Plotted Against Specific Gravity.

temperature. Typical values for the three temperatures are 0°, 15°, 25°. If brine is used, the ammonia will work between

0° and 90°. With direct expansion, it might have worked between 15° and 90°. The latter range gives a more efficient cycle. Direct expansion is greatly to be preferred on economical grounds for ice-making or very low temperature work. Brine circulation gives opportunity for storage of "cold" and no harm is done if the compressor has to be shut down for a short period.

In INDIRECT REFRIGERATION, air is used instead of brine. Air is blown over coils carrying the refrigerating fluid, and the chilled air then passes to the room to be cooled.

Fig. 37 gives the properties of the two types of brine used. Calcium chloride permits of lower temperatures than salt brine, and is less corrosive.

Prob. 225. In Prob. 224, the machine as actually installed developed 98 tons capacity and the compressor cylinder indicated 107 hp. while the Ihp. of the steam engine driving the compressor was 133.33. State the relative efficiency and the mechanical efficiency. (Ans., 0.72, 0.80.) If the actual indicated thermal efficiency of the steam engine was 0.10, what was the efficiency from steam at the throttle to refrigeration? (Ans., 0.38.)

Prob. 226. What is the Carnot efficiency of refrigeration between 0° and 90° F.? (Ans., 5.1.) Between 15° and 90°? (Ans. 6.3.)

Prob. 227. The machine in Prob. 221 is used for indirect (air-blast) refrigeration. The air is cooled at constant pressure from 70° to 10°. Ignoring radiation loss, what weight of air is circulated per min.? (Ans., 1400 lb.)

Prob. 228. A calcium chloride solution has a freezing point of − 20°. What is its strength? (Ans., 24.4 per cent.) What is its specific gravity? (Ans., 1.217.) Its specific heat at 5° F.? (Ans., 0.688.) If with a 100 ton machine its temperature rises from − 15° to 25° F., what weight of solution must be pumped per min.? (Ans., 726 lb.)

116. **Tonnage Rating.** The 100 ton machine of Prob. 223 has a displacement of 454 cu. ft. per min., or $4.54 \times 1728 = 7840$

cu. in. per ton per min. This figure (the SPECIFIC DISPLACEMENT) depends on the clearance, the limiting temperatures and the dryness at the beginning of compression. There is a tendency to standardize 0° and 95½° as limits for rating. Under these conditions a specific displacement proposed is 5.52 cu. ft. or 9570 cu. in.

Prob. 229. In Art. 116, if clearance is 5 per cent., and temperature limits are 0° and 95½°, what value of x_b would give a specific displacement of 9570? (Ans., 0.94.)

117. **Dry Compression.** In Prob. 229, the vapor is nearly dry at the BEGINNING of compression. Inspection of Fig. 36 will show that it is bound to become somewhat superheated during adiabatic compression to 95½°. Such operation is described as "dry compression," in distinction to "wet compression" heretofore described. Wet compression leads to the more efficient cycle, especially if x_b is so adjusted that $x_c = 1.0$. This is a condition difficult to maintain, and it is obviously NOT the condition which gives maximum refrigeration per lb. of fluid. Such maximum refrigeration is obtained (without allowing the fluid temperature to rise in the cold room) when $x_b = 1.0$. In order to maintain wet compression under this condition, something must be done to withdraw heat during compression. This makes the path depart from the adiabatic, swinging it to the left as it proceeds upward. Jackets can be used to accomplish this result, as in air compressors. Multi-stage operation is occasionally practiced. The injection of cold oil or of liquid ammonia into the cylinder is frequently adopted. In any case, the rate of flow of fluid should be increased as soon as the compressor discharge becomes noticeably warm. This indicates (with normal values of p_c) the existence of some superheat. Increasing the flow of fluid will decrease x_b and reduce or eliminate the superheat.

Prob. 230. With 0° and 90° limits, find x_b when $x_c = 1.0$. (Ans., 0.88.)

Prob. 231. While steadily giving 100 tons of refrigeration, the machine in Prob. 217 is speeded up so that the weight of fluid circulated per min. is increased 10 per cent. What change occurs in x_b? (Ans., it is reduced from 0.87 to 0.80.)

118. **Ideal Cycle with Superheat.** In ideal dry compression without heat removal, we have such a cycle as *ejkld*, Fig. 36. The point k is not located to scale. The line lk is a line of constant pressure. We now have, following Art. 114,

Refrigeration done $= L_{fj}(1 - x_f)$,

Heat rejected at condenser $= H_k' - h_d$,

Work expended $= H_k' - H_j$,

$$\text{Efficiency} = \frac{L_{fj}(1 - x_f)}{H_k' - H_j},$$

$$\text{Mean effective pressure} = p_m = \frac{5.403(H_k' - H_j)}{v_j},$$

Volumetric efficiency as in Art. 115.

The value of x_f is found as before, in Art. 113. That of H_k' is best found from a total heat-entropy diagram, Fig. 38. The entropy and pressure at k are known. The symbol H_j means the total heat of dry vapor at j.

Thus take limits of 0° and 70°. From Prob. 216, $x_f = 0.132$. Then

Refrigeration done $= 572.2\ (1 - 0.132) = 497$,

Pressure at $k = 129.2$,

Entropy at $k = (n_s)_{00} = 1.1741$,

H_k' from Fig. 38 $= 630$,

Heat rejected $= 630 - 42.1 = 587.9$,

Work $= 630 - 538.5 = 91.5$,

Efficiency $= 497 \div 91.5 = 5.45$,

$p_m = (5.403 \times 91.5) \div 9.19 = 54$.

The efficiency is less than in Prob. 220 and the mean effective pressure is necessarily greater. For 100 tons capacity, 40.25 lb. of fluid must be circulated per min. This is of course less than in Prob. 221. The volume of fluid per min. is $40.25 \times 9.19 = 370$ cu. ft., and with volumetric efficiency $= 0.834$ as in Prob.

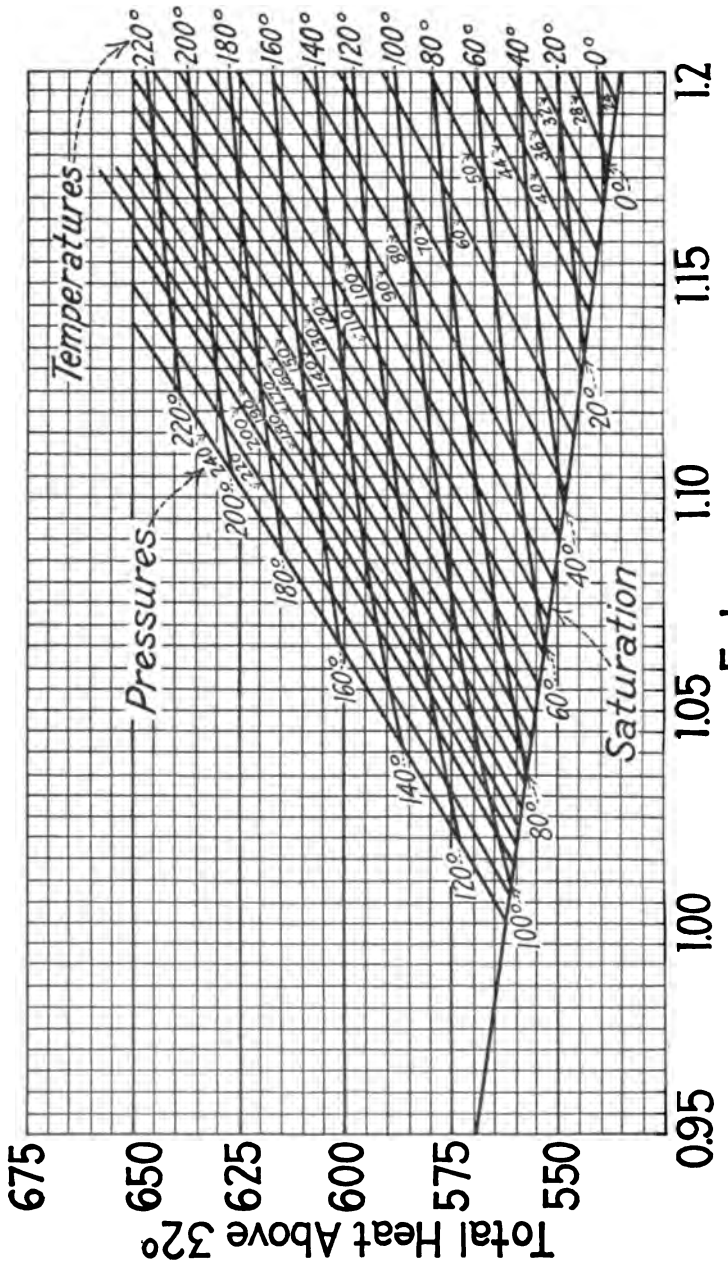

FIG. 38. Total Heat-Entropy Diagram for Superheated Ammonia. (Plotted from Goodenough's "Properties of Steam and Ammonia.")

223, the specific displacement is $(3.7 \times 1728) \div 0.834 = 7700$ cu. in., a value not notably different from that realized in wet compression.

Prob. 232. Find the ideal efficiency with dry compression from 0° to 90°. (Ans., 4.12.)

119. **Carbon Dioxide Machine.** The temperature-entropy diagram, Fig. 39, is plotted with lines of constant pressure and constant superheat in the superheat region. Dry compression

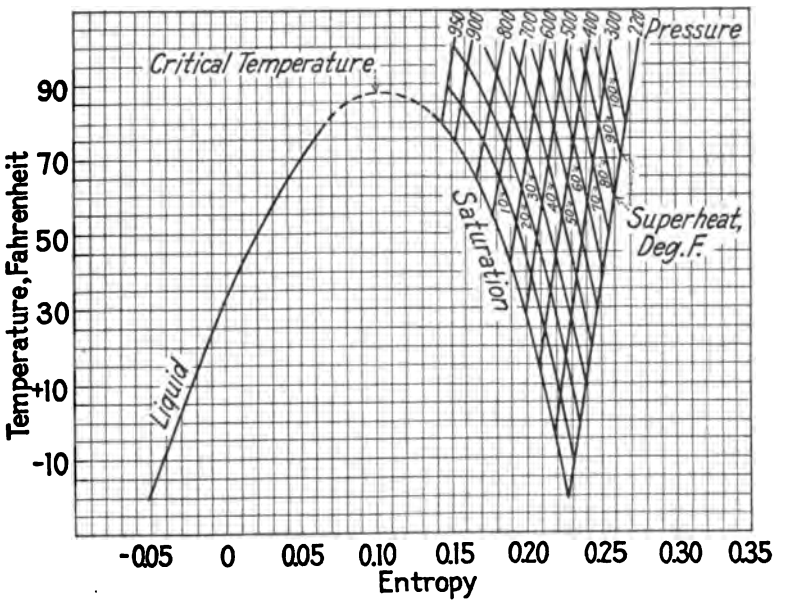

FIG. 39. Temperature-Entropy Diagram for Carbon Dioxide.

is common, but leads to low efficiency. Adiabatic compression from a dry state at 0° to the pressure of 900 lb. (corresponding roughly with a saturation temperature of 75°) would put the final temperature far above 100°. In the tropics, the vapor could not then be condensed. We must in practice in any latitude either compress from an initially wet condition, or remove heat during compression, or be content with comparatively low efficiency.

Properties of Saturated Carbon Dioxide Vapor

t, Temperature, Deg. F.	p, Pressure, Lb. per Sq. In.	v, Volume, Cu. Ft. per Lb.	v_w, Volume of Liquid, Cu. Ft. per Lb.	h	L	H	r	n_w	n_s	n_g
0	313.5	0.2661	0.016298	−15.5	114.3	98.8	99.81	−0.03278	0.2494	0.21662
75	902.0	0.0708	0.022232	29.4	50.8	80.2	42.69	0.05718	0.09495	0.15213

In the first case, the method of analysis is precisely as in Arts. 113, 114. In calculating mean effective pressure and displacement, however, it should be noted that the values of v and v_w are too nearly the same to permit of the approximation heretofore used (Arts. 74, 115) for the specific volume at the beginning of compression. That volume is for vapor at 0° F., 0.90 dry, for example, NOT 0.90×0.2661, but $0.90\ (0.2661 - 0.016298) + 0.016298 = 0.2418$.

Prob. 233. With CO_2 between 0° and 75°, the vapor is exactly dry at the end of adiabatic compression. What was its dryness at the beginning of compression? (Ans., 0.743.)

Prob. 234. In Prob. 233, what was the dryness at the expansion valve outlet? (Ans., 0.39.) Compute the refrigeration per lb. of fluid. (Ans., 40 B.t.u.) How much heat is rejected at the condenser? (Ans., 51 B.t.u. per lb.) How much work is done? (Ans., 10.5 B.t.u. per lb.) What is the ideal efficiency? (Ans., 3.83.) What is the specific volume at the beginning of compression? (Ans., 0.202.) What is the volumetric efficiency if clearance is 4 per cent.? (Ans., 0.9248.) What is the displacement per lb. of fluid? (Ans., 0.218 cu. ft.) What is the specific displacement — cu. in. per min. per ton — for the conditions assigned? (Ans., 1870.)

120. **Carbon Dioxide Cycle with Heat Removal.** Suppose heat to be removed during compression at such rate as to make the vapor dry at both the beginning and end of compression. The cycle is represented diagramatically (not to scale) by *ejld*,

Fig. 36. Assume the compression path jl to be a polytropic. Then $p_l v_l^a = p_j v_j^a$, and for the 0°–75° limits, the exponent is

$$a = \frac{\log p_l - \log p_j}{\log v_j - \log v_l} = \frac{\log 902 - \log 313.5}{\log 0.2661 - \log 0.0708}$$

$$= \frac{2.955 - 2.496}{-0.5744 + 1.15} = 0.798.$$

Then the external work done under this curve jl is

$$W_{jl} = \frac{p_l v_l - p_j v_j}{a - 1} = \frac{(902 \times 0.0708) - (313.5 \times 0.2661)}{-0.202}$$

$$\times \frac{144}{778} = 17.9 \text{ B.t.u.}$$

The external work of the whole cycle is

$$\Sigma W = W_{ed} + W_{dl} + W_{lj} - W_{je}$$

$$= \frac{(pv_w)_d}{5.403} + \frac{p_l(v - v_w)_l}{5.403} + 17.9 - \frac{p_j(v - v_w)_j}{5.403}$$

$$= \frac{(pv)_l}{5.403} + 17.9 - \frac{p_j(v - v_w)_j}{5.403} = \frac{902 \times 0.0708}{5.403}$$

$$+ 17.9 - \frac{313.5(0.2661 - 0.0163)}{5.403}$$

$$= 11.8 + 17.9 - 14.5 = 15.2 \text{ B.t.u.}$$

The refrigeration done is $L_j(1 - x_f)$. The heat rejection at the condenser is L_l. The specific volume at the beginning of compression is the tabular volume v_j. Since $\Sigma H = \Sigma W$,

$$H_l + H_{lj} - H_j = 15.2 = 80.2 + H_{lj} - 98.8$$

$$H_{lj} = 33.8 \text{ B.t.u. per lb. of fluid.}$$

This is the quantity of heat to be removed during compression.

Prob. 235. In the foregoing, find refrigeration and heat rejection per lb. of fluid, and the ideal efficiency. (Ans., 69.6, 50.8, B.t.u.: efficiency = 4.59.)

Prob. 236. The machine in the foregoing is of 100 ton capacity, double-acting, at 240 ft. piston speed and 5 per cent.

clearance. Find cylinder diameter, ideal hp. and ideal ice-melting effect per Ihp.-hr. (Ans., 8 in., 103 hp., 81 lb.)

Prob. 237. In Probs. 235–236, 33.8 B.t.u. must be removed per lb. of fluid during compression. If this is accomplished by injecting liquid at 0° into the cylinder, what weight of fluid must be thus supplied per min., the injected liquid being completely vaporized at 75°? (Ans., 102 lb.)

121. **Comments.** This last type of cycle requires less power for a given amount of refrigeration, and is consequently more efficient, than the cycle with adiabatic compression: but it requires jacketing or liquid injection. Carbon dioxide gives lower thermal efficiencies than ammonia, but much higher efficiencies than air. On account of its low specific volume (high density) the displacement is less in proportion to refrigerating capacity than is the case with ammonia. The very high pressures reached limit the application of carbon dioxide to cylinders of small diameter. Hence large machines rarely use this fluid. Leakage is more probable than with ammonia, and is not easy to detect. The fluid is usually cheaper than ammonia. For extremely low temperatures, carbon dioxide can be used without maintaining a vacuum between the compressor and the expansion valve.

122. **Sulphur Dioxide.** This fluid is occasionally employed in marine service. The following values are given by Zeuner:

Properties of Saturated Sulphur Dioxide

t, Temperature Fahrenheit	p, Pressure, Lb. per Sq. In.	v, Volume, Cu. Ft. per Lb.	h	L	H	r	s_w	s	s_s
−4	9.272	8.051	−11.216	171	159.784	157.111	−0.0237	0.3755	0.3518
5	11.756	6.486	− 8.449	169.745	161.296	155.563	−0.0177	0.3655	0.3478
77	56.386	1.445	14.582	148.775	163.357	133.718	0.0284	0.2773	0.3057
86	66.359	1.220	17.572	144.787	162.359	129.827	0.0339	0.2655	0.2994
95	77.630	1.036	20.588	140.495	161.083	125.656	0.0394	0.2534	0.2928

Prob. 238. What property, in the above table, varies in an unusual way?

CHAPTER VII

HEAT TRANSFER APPLIANCES

123. **Mixtures.** Mixtures of SIMPLE SUBSTANCES have been considered in Art. 3. For these, or for VAPORS, ignoring radiation,

$$\text{loss of heat} = \text{gain of heat},$$
$$w_a(Q_a' - Q_a) = w_b(Q_b' - Q_b),$$

where w = weight in lb., Q = heat content in B.t.u., and primes denote the high temperature conditions. In actual mixing, $t_b' = t_a$: i.e., the final temperatures of the two fluids are the same. For simple substances, $w(Q' - Q)$ is best written, $ws(t' - t)$, where s = specific heat. For vapors,

$Q = h$ at the liquid condition,
$Q = h + xL$ when wet,
$Q = H$ when dry,
$Q = H + k(t_{sup} - t_{sat})$ when superheated.

Prob. 239. Ammonia at 199 lb. pressure, superheated 220°, is cooled, condensed and further cooled until its temperature is 70° F. See Fig. 38. How much heat is emitted per lb.? (Ans., 598.9 B.t.u.)

Prob. 240. An injector is supplied with dry steam at 200 lb. pressure and water at 90° F. The two fluids mix and the resulting temperature is 170.06° F. What weight of water was mixed with 1 lb. of steam? (Ans., 13.3 lb.)*

Prob. 241. In a jet condenser, 75.07 lb. of water are employed per lb. of steam. Water enters at 80° F. Steam enters at 2 lb. pressure, 0.70 dry. What is the temperature of the discharged mixture? (Ans., 90° F.)

124. **Temperatures in Surface Transfer.** Transfers through surfaces (like the metal wall of a condenser tube) are more

* See Art. 134.

complicated. There are four temperatures to be considered; the inlet and outlet temperatures of both the heat-emitting and the heat-receiving fluids. The rate of transfer depends not only on the VALUES of these four temperatures, but also, so to speak, on their ARRANGEMENT. Take the case of an economizer—a group of tubes containing feed water, placed in the path of the combustion gases from a boiler to its stack. The water moves vertically through the tubes, but it also moves (considering the whole system of tubes) in a direction which may be either TOWARD the boiler or AWAY from it.

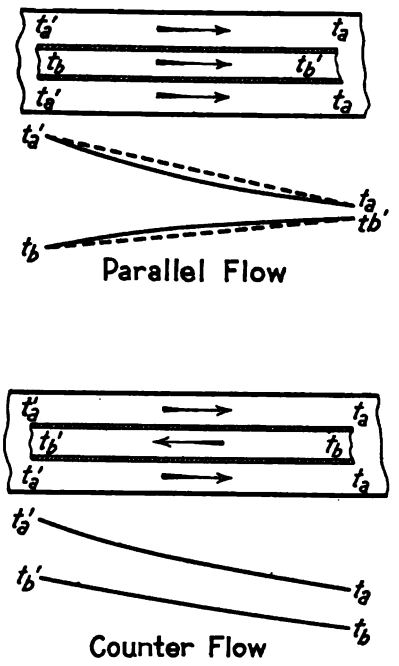

FIG. 40. Two Cases of Heat Transfer.

The two cases thus suggested are represented diagrammatically in Fig. 40. In PARALLEL FLOW, the two fluids move in THE SAME general direction. In COUNTER FLOW, they move in OPPOSITE directions. In many devices, the direction of flow is uncertain or indeterminate. Such is the case with a steam radiator, for instance.

The rate of heat flow is generally proportional to the temperature difference, that is, to the difference between the temperatures of hot and cold fluids. The total heat transfer is proportional to the MEAN TEMPERATURE DIFFERENCE, t_m. This is the average ordinate or vertical distance between the two curves of Fig. 40, (upper diagram). It is NOT the arithmetical AVERAGE temperature difference, which would be the average vertical distance between the two dotted straight lines. For parallel flow, t_m is

obviously LESS than the arithmetical average. Its value for parallel flow or for cases where one fluid remains at constant temperature, or for cases in which there is no determinate flow, is given by

$$t_m = \frac{t_a' - t_b - (t_a - t_b')}{\log_e \frac{t_a' - t_b}{t_a - t_b'}}. \tag{68}$$

For counter flow, it is

$$t_m = \frac{t_a' - t_b' - (t_a - t_b)}{\log_e \frac{t_a' - t_b'}{t_a - t_b}}, \tag{68a}$$

symbols being as in Art. 123. The derivation of these formulas furnishes an interesting exercise in calculus.

In parallel flow, t_b' approaches t_a as a maximum. In counter flow, it approaches the higher value t_a'. Hence counter flow permits of higher ultimate temperatures of the heat-receiving fluid. On the other hand t_b' actually becomes equal to t_a' or t_a only when the surface is infinite. A finite difference of temperature always exists in practice at the outflowing point of the colder fluid (i.e., the right-hand end for parallel flow and the left-hand end for counter flow, in Fig. 40). With this difference fixed, t_m is greater for parallel flow. The quantity of heat transmitted is proportional to t_m. Hence with a fixed temperature difference at the outflow of the cooler fluid, and when a given quantity of heat is to be transmitted, parallel flow is more economical of transmitting surface.

Prob. 242. One fluid cools from 100° to 60° while the other is heated from 30° to 50°. What is the arithmetical average temperature difference? (Ans., 40°.) State the value of t_m for parallel flow. (Ans., 30.9.) The direction of flow of the cooler fluid is now reversed and its final temperature changes to 68°. Find t_m. (Ans., 30.9.) Compare the heat transfers in the two cases. (Ans., they are equal.) Compare temperature differences at cold fluid outlet. (Ans., parallel flow, 10°, counter flow 32°.)

Prob. 243. Should parallel flow or counter flow be used in the condenser of a CO_2 refrigerating machine? Why?

125. **Coefficient of Transmission.** The general equation of transmission is

$$w_a(Q_a' - Q_a) = w_b(Q_b' - Q_b) = \rho A t_m, \qquad (69)$$

where weights are in LB. PER HR., A = surface in sq. ft., and ρ = COEFFICIENT OF TRANSMISSION. Obviously, ρ indicates THE QUANTITY OF HEAT (B.t.u.) TRANSMITTED PER HR. PER DEGREE OF MEAN TEMPERATURE DIFFERENCE BY EACH SQ. FT. OF SURFACE. Few factors used in engineering are more variable than ρ. It depends on the two fluids, their condition and purity: and on the cleanliness and material of the transmitting surface. It is highest for transfer between liquids and wet vapors, as in condensers and feed water heaters: and lowest for transfer involving dry gases, as in superheaters and economizers. Besides the factors already mentioned, ρ varies notably with some power (0.3 to 0.6) of the VELOCITY of fluid, or more strictly with the MASS-FLOW (velocity $\times$ density). The values of ρ given in the accompanying table are merely rough averages.

Prob. 244. A fan discharges 100 lb. of air per min. across coils through which water is being pumped. The water cools from 197.75° to 170.06°. The air is heated from 70° to 130°. What weight of water is pumped per min.? What amount of coil surface is necessary? Note: direction of flow is indeterminate, hence use Equation (68). (Ans., 51 lb., 113 sq. ft.)

Prob. 245. A feed water heater uses auxiliary exhaust steam, 0.70 dry at 25 lb. pressure, amounting to 10 000 lb. per hr. Boiler feed water enters at 90°, its weight being 150 000 lb. per hr. The condensed steam is 10° warmer than the outlet feed water. What will be the outlet feed temperature? If counter flow is used, find t_m and the necessary surface. (Ans., 140°, 274 sq. ft.)

Prob. 246. In Art. 47, 130 lb. of air are cooled 244° F., per minute, in the intercooler. Assume indeterminate flow, water entering at 70° and leaving at 110°. Find t_m, and the necessary intercooler surface if $\rho = 3$. (Ans., $t_m = 199°$, surface = 755 sq. ft.)

Operating Data, Heat Transfer Appliances

Device	Value of ρ	Heat Emission				Heat Reception			
		Initial		Final		Initial		Final	
		Substance	Condition	Substance	Condition	Substance	Condition	Substance	Condition
Steam radiator	1.8	Wet steam	p = 12 to 100 x = 0.7 to 1.0	Water	t = 150 to 200	Air	t = 70	Air	t = 70
Hot water radiator	1.8	Water	t = 160 to 220	Water	t = 140 to 200	Air	t = 70	Air	t = 70
Fan system heater	10*	Steam	p = 16	Water	t = 200	Air	t = 0 to 40	Air	t = 80 to 130
Refrigerating coils	2	Air	t = 0 to 40	Air	t = 0 to 40	Ammonia vapor	x = 0.10 to 0.20 t = −20 to +20	Ammonia vapor	x = 0.90 to 1.0 t = −20 to +20
Refrigerating coils	2	Air	t = 0 to 40	Air	t = 0 to 40	Brine	t = −20 to +20	Brine	t = 0 to 30
Intercooler	3	Air	t = 200 to 360	Air	t = 90 to 200	Water	t = 60 to 75	Water	t = 80 to 140
Air cooler (dense air ice machine)	3	Air	t = 250 to 350	Air	t = 95 to 125	Water	t = 60 to 75	Water	t = 80 to 90
Economizer	4	Gas	t = 500 to 700	Gas	t = 320 to 450	Water	t = 90 to 120	Water	t = 190 to 290
Superheater	3	Gas	t = 1000 to 1400	Gas	t = 800 to 1100	Steam	p = 150 to 250	Steam	p = 150 to 250 t = 500 to 650
Boiler	10	Gas	t = 2400 to 3000	Gas	t = 500 to 700	H_2O	p = 100 to 300	H_2O	p = 100 to 300
Feed water heater	350	Wet steam	p = 15 to 25 x = 0.7 to 0.9	Water	t = 200 to 210	Water	t = 60 to 120	Water	t = 160 to 215
Surface condenser	350	Wet steam	p = ½ to 2 x = 0.7 to 0.9	Water	t = 85 to 115	Water	t = 55 to 70	Water	t = 80 to 110
Brine cooler	50†	Brine	t = 10 to 40	Brine	t = −20 to +20	Ammonia vapor	x = 0.10 to 0.20 t = −30 to +10	Ammonia vapor	x = 0.90 to 1.0 t = −30 to +10
Ammonia condenser	250	Ammonia vapor	x = 1.0 t = 75 to 100	Ammonia liquid	t = 75 to 95	Water	t = 55 to 70	Water	t = 70 to 90
Receiver, comp'd eng.	400	Steam	x = 0.95 to 1.0 p = 125 to 250	Water	t = 300 to 400	Steam	x = 0.70 to 0.90 p = 15 to 40	Steam	x = 0.90 to sup. p = 15 to 40
Evaporator (single stage)	300	Steam	x = 0.95 to 1.0 p = 40 to 90	Water	t = 250 to 300	Water	t = 70 to 150	Steam	p = 20 to 50 x = 0.97 to 0.99

* Air blown at high velocity over coils.

† Can be increased by improving brine circulation.

126. **The Steam Boiler.** If the potential heat (heat of combustion) of the coal burned is B_1, the heat evolved in the furnace is always some less quantity B_2, the quotient B_2/B_1 being e_F, the FURNACE EFFICIENCY. The heat evolved, B_2, is AVAILABLE for transmission through the heating surface to the water and steam. Only a part, B_3, is so transmitted, the remainder $(B_2 - B_3)$ being lost up the stack. Then $B_3/B_2 = e_s$ is the SURFACE EFFICIENCY or transmissive efficiency and $B_3/B_1 = e_F e_s$ is the efficiency of the whole boiler.

For uniform coal, B_1 varies in direct proportion to the amount of coal burned. Under good control, e_F is about constant. Hence with such good control B_2 is directly proportional to the coal burned. The heat B_2 exists in the form of a body of hot gas, and is proportional to the weight and temperature of gas. It is numerically equal to the product of weight of gas by its mean specific heat and by the difference between fire-room temperature and furnace temperature. The furnace temperature and the mean temperature difference between gas and steam are scarcely affected by the weight of coal burned. The weight of gas is (or should be, in good operation) directly proportional to the weight of coal.

The heat transmitted is $B_3 = \rho A t_m$. In a given boiler, A is constant. Under good operation, as above described, t_m is constant. Hence B_3 varies directly with ρ. Accept the mass-flow theory, i.e., ρ varies directly with the weight of gas, hence with the weight of fuel. Then B_3 varies directly with the weight of fuel. If e_F varies, B_3 nevertheless varies directly with B_2.

This analysis loses sight of the fact that part of the transfer in a boiler is by RADIANT HEAT, with which the transmission varies not as the first power, but as the FOURTH POWER of the mean temperature difference. However, if t_m near the furnace is constant, the radiant heat emission is constant. Hence the BALANCE of heat transmitted is directly proportional to B_2, or, at constant furnace efficiency, to B_1.

This deduction is thoroughly confirmed by the straight line law illustrated in Fig. 41. The radiant heat emission is constant,

and does not depend on the amount of coal burned. As a mathematical fiction, it may be regarded as represented by the intercept on the vertical axis (*oa*, for the Denver). The BALANCE of the heat is directly proportional to B_1. Hence e_s decreases

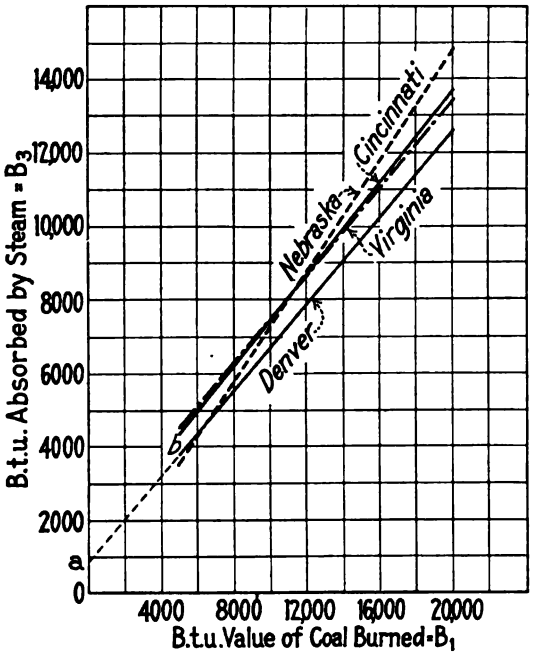

FIG. 41. Navy Department Tests on Water-Tube Boilers—Quantities are per Square Foot of Heating Surface per Hour.

with forcing, but more and more slowly, and the limit of boiler capacity is not easily reached. At very high rates of driving, e_r decreases.

If tests were carried down to zero weight of coal burned, the lines could not, obviously, continue straight. This would imply NEGATIVE radiant heat for the Nebraska. The imaginary straight portion (*ab* for the Denver) may, however, be utilized in calculation. See Probs. 247–250.

A boiler "horse power" is equivalent to the transmission of $34.5 \times 970.4 = 33\,479$ B.t.u. per hr., or the evaporation of $34\frac{1}{2}$ lb. of water per hr. from and at 212° F.

Prob. 247. For the Denver, in Fig. 41, how much heat was transmitted by pure radiation? (Ans., 860 B.t.u. per sq. ft. of heating surface per hr.)

Prob. 248. When 20 000 B.t.u. coal value was supplied per sq. ft. of heating surface per hr., in the Denver, how much heat was transferred by radiation? (Ans., 860 B.t.u.) By ordinary transmission? (Ans., 11 800 B.t.u.) What was the boiler efficiency? (Ans., 0.633.) What was the hp. per sq. ft. of heating surface? (Ans., 0.377.)

Prob. 249. State the boiler efficiency and hp. per sq. ft. of heating surface for the Denver when the coal equivalent is $B_1 = 10\,000$ B.t.u. per hr. (Ans., 0.674 efficiency, 0.202 hp.)

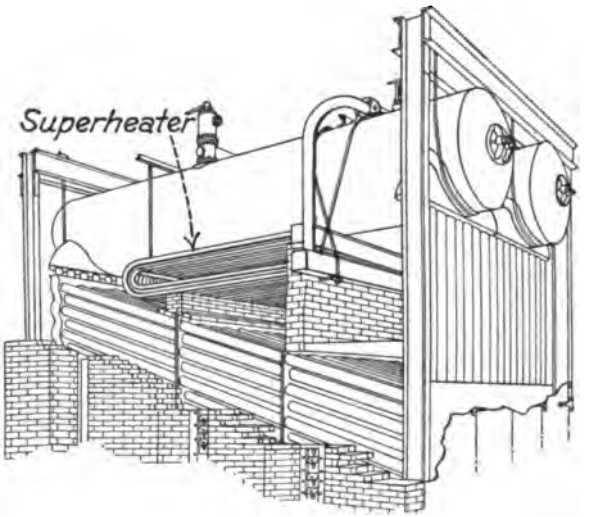

FIG. 42. Babcock and Wilcox Stationary Boiler with Superheater.

Prob. 250. If $t_m = 1200$, find ρ in Probs. 248, 249. Note: ρ applies to ordinary transmission only. (Ans., 9.83 and 4.90.)

127. **Superheaters.** Superheaters (Fig. 42) are usually placed between the first and second "passes" of a water tube boiler, where the gas temperature is well above 1000°. The transfer equation is

$$w_s k_s(t_s' - t_s) = w_g k_g(t_g' - t_g) = \rho A t_m, \qquad (70)$$

where w_s = lb. of steam per hr.

k_s = specific heat of superheated steam between t_s and t_s' at the given pressure.

t_s' = temperature of steam after superheating,

t_s = temperature of saturated steam at the given pressure,

w_g = lb. of combustion gases passing the superheater per hr.,

k_g = specific heat at constant pressure of combustion gases, say 0.24,

t_g' = temperature of gases approaching superheater,

t_g = temperature of gases leaving superheater,

ρ = coefficient of transmission, around 3,

t_m = mean temperature difference,

A = superheater surface, sq. ft.

In very good operation, w_g will vary directly with w_s, the ratio w_g/w_s being from 1.6 to 2.2. Hence

$$\frac{t_s' - t_s}{t_g' - t_g} = \frac{2 \times 0.24}{0.48} = 1$$

when $w_g/w_s = 2$ and $k_s = 0.48$ (rough average values). Thus the rise in temperature of the steam is about equal to the fall in temperature of the gas. As with boilers, ρ increases in direct proportion when w_g increases: i.e., when THE BOILER is forced. Under these conditions, the equations can be satisfied only when the temperature ranges remain unchanged, because t_m cannot increase when either temperature range increases. In practice, the ratios w_g/w_s and ρ/w_g both vary somewhat and the amount of superheat varies in an irregular way with the rate of driving.

PARALLEL FLOW should be used in superheaters to decrease the necessary surface.

Prob. 251. A boiler of 1000 hp. uses feed water at 213° and generates dry steam at 200 lb. pressure. What weight of steam is produced per hr.? (Ans., 33 000 lb.)

Prob. 252. In Prob. 251, it is required to superheat this steam 200°. Take $k_s = 0.548$. The boiler evaporates 8.5 lb. of water per lb. of coal and the weight of gas is 17 lb. per lb. of coal. The gas after passing the superheater is at 1000° F. What is its temperature at entrance to the superheater? (Ans., 1230° F.) What is the mean temperature difference? (Ans., 610° F.) The superheater surface? (Ans., 1970 sq. ft.) If the coal used contains 11 267 B.t.u. per lb., what is the combined efficiency of boiler and superheater? (Ans., 0.85.) Note: efficiencies above 0.80 are rarely attained in practice.

Prob. 253. In Prob. 252, 10 per cent. of the heat is transmitted by pure radiation. Given $t_m = 1200$, what should be the heating surface (not including superheating surface) of the boiler? (Ans., 2500 sq. ft.)

128. **Economizer.** The definite equation is (see Art. 124),

$$w_w(t_w' - t_w) = 0.24w_g(t_g' - t_g) = 4At_m, \qquad (71)$$

where w_g/w_w, as with the superheater, tends to remain around 2 in the best operation. Then the water is warmed about half as many degrees as the gas is cooled. COUNTER FLOW gives the highest final water temperatures, but economizer surface is costly and PARALLEL FLOW may therefore be used.

Prob. 254. Boiler feed water amounting to 10 000 lb. per hr. is to be heated in an economizer. Water enters at 90°, gas at 500° and the ratio of weights of gas and water is 5.0. With parallel flow, the water leaves at 234°. Find the outlet gas temperature and the surface. What is the value of $(t_a - t_b')$, Fig. 40 ? (Ans., $t_g = 380°$ F., $A = 1400$ sq. ft., $(t_a - t_b') = 146$.

Prob. 255. If $w_w = 5454.54$, $t_w = 90$, $w_g = 5w_w$, $t_g' = 500$, with COUNTER FLOW, find the outlet gas temperature and the surface when $t_w' = 354°$. (Ans., $t_g = 280°$, $A = 2130$ sq. ft.) Compare costs of installation and saving of heat in Probs. 254 and 255. (Ans., the temperature differences at water outlets are the same; the quantities of heat transmitted are the same; the counter flow economizer is 52 per cent. larger.)

129. **Surface Condenser, Feed Water Heater.** In surface condensers (Fig. 43) and feed water heaters (Fig. 44), some of the factors affecting the value of ρ have been established. For usual tubes when clean, $\rho = 400\sqrt{u}$, where u = water velocity, ft. per sec. Very badly fouled tubes will give half this value of ρ.

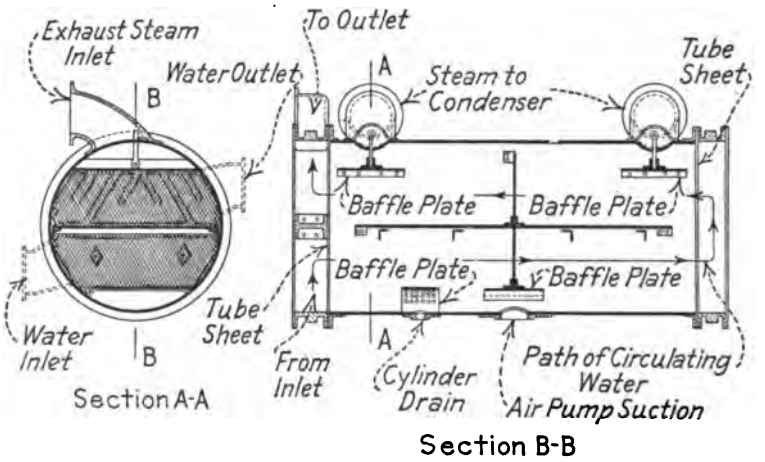

FIG. 43. Surface Condenser Showing Path of Water and Steam and General Arrangement of Baffle Plates and Tubes.

VULCANIZING (coating) admiralty tubes decreases ρ nearly 85 per cent. If there is much air in the steam to be condensed, ρ will be reduced. Air may enter through leaks in vacuum piping. Fresh feed water also carries air. High water velocities increase ρ and therefore decrease the necessary size of heater or condenser. They also increase the power consumption of the pump which handles the water. This last point is not of much importance in condensers, where circulating water pressures are low: but the water passing through feed water heaters is under full boiler pressure and excess pressure should be avoided. With high water velocities, excess pressures are necessary to overcome friction. For feed water heaters, use $u = 6$ or 7.

For condenser or heater, with clean tubes of usual materials,

$$w_s(h_1 + x_1L_1 - h_2) = w_w(t_4 - t_3) = 400\sqrt{u}At_m, \qquad (72)$$

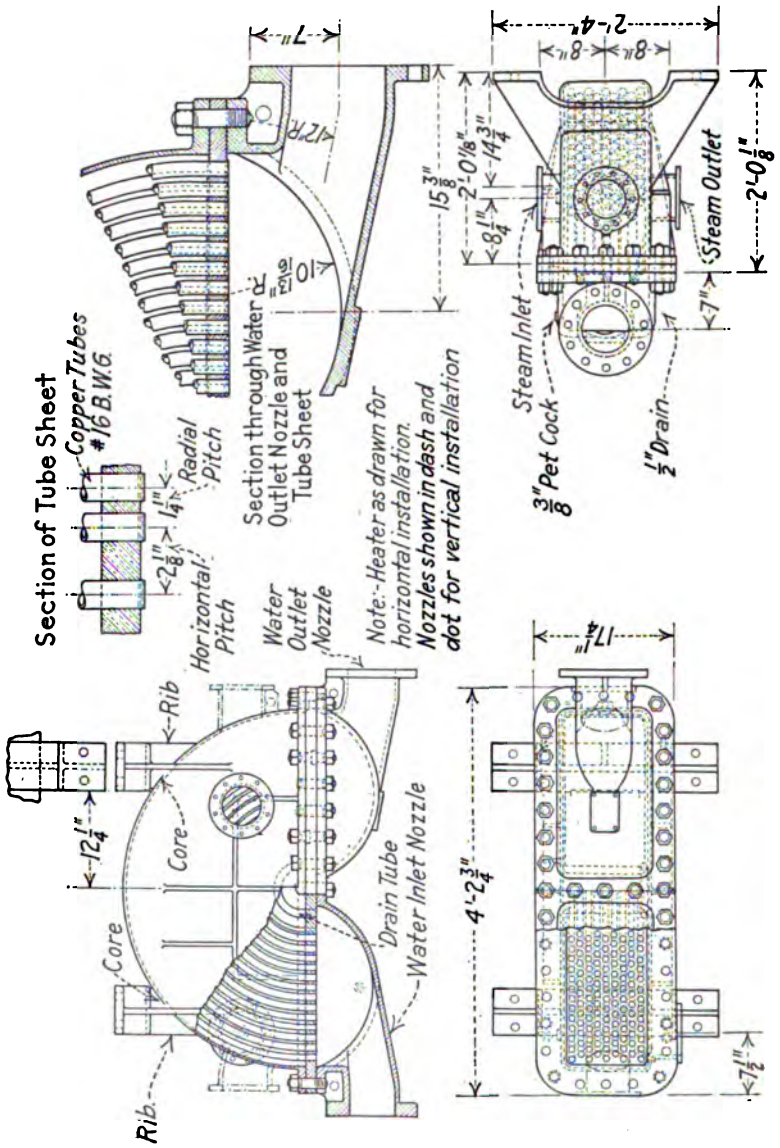

FIG. 44. Marine Feed Water Heater.

where $_1$ and $_2$ are initial and final steam conditions, $_3$ and $_4$ initial and final water conditions. Flow is usually indeterminate, hence Equation (68) is used for t_m: this equation becoming, in present symbols,

$$t_m = \frac{t_1 - t_3 - (t_2 - t_4)}{\log_e \frac{t_1 - t_3}{t_2 - t_4}}.$$

Prob. 256. A condenser handles 17 400 lb. of steam per hr. at 0.696 lb. pressure, 0.85 dry. The condensed steam is discharged at 80°. The circulating water enters at 60° and leaves at 74°. Find the weight of water. (Ans., 1 113 000 lb. per hr.)

Prob. 257. The condenser in Prob. 256 has 600 tubes in each "pass"—see Fig. 42. These each have an internal cross-sectional area of 0.219 sq. in. If the weight of water is 64 lb. per cu. ft., find the water velocity through the tubes. (Ans., 5.3 ft. per sec.)

Prob. 258. In the foregoing, find the values of ρ, t_m and A. (Ans., $\rho = 922$, $t_m = 14.9$, $A = 1140$.)

Prob. 259. In Probs. 256–258, each lineal ft. of tube gives 0.1635 sq. ft. of surface. There are two passes. What is the length of tubes? (Ans., 5.8 ft.)

In FEED WATER HEATERS, ρ is sometimes increased by the use of retarders or by employing corrugated tubes. Sometimes a double corrugated tube is used, the water passing through the narrow space between the inner and outer walls. The object of increasing ρ is to decrease surface and thereby to save space and cost. Many special forms of heater increase ρ without decreasing total bulk or cost, because the surface is of such shape as to be of large bulk per unit of transmitting area. Condensers generally have plain straight tubes.

130. **Evaporator.** Essentially, an evaporator is a shell containing salt water in which there is immersed a coil containing steam. Pure steam is driven off from the sea water and later condensed, thus furnishing a supply for boiler feed and domestic

purposes. Fig. 45 shows considerable elaboration on this. The salt feed is first used as cooling water in a small condenser or "distiller." The bulk of it is then wasted, but a part feeds the evaporator shell. Its temperature having been somewhat raised

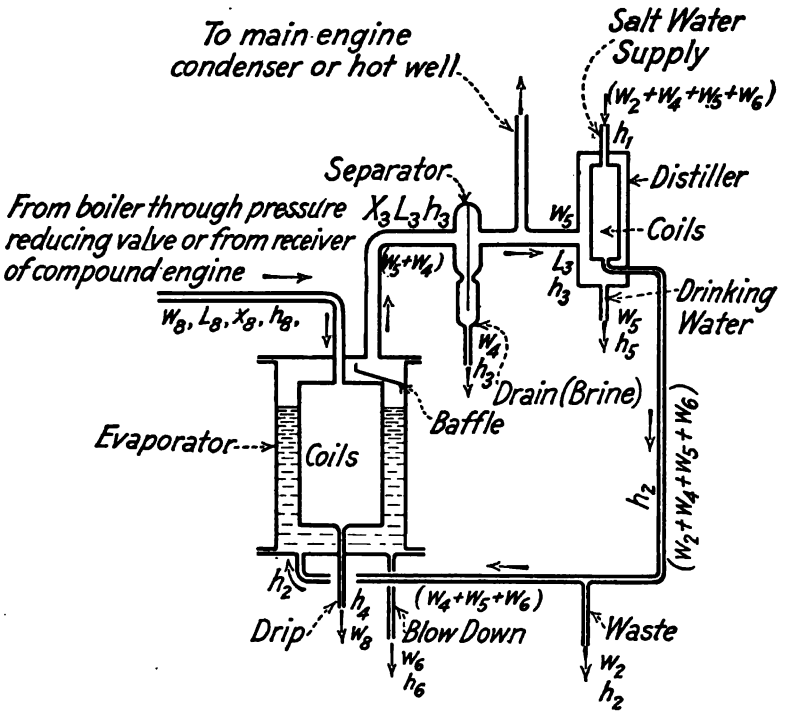

FIG. 45. Evaporator.

at the distiller, less heat will now be required to evaporate than would otherwise be needed. In the evaporator the salt water is boiled: and, if some of the coil is unsubmerged, slightly superheated. As the sea water is vaporized, the salt left behind continually strengthens the brine in the evaporator shell. Hence the contents must be frequently or steadily blown down to keep the salinity within reasonable limits.

The vapor leaving the evaporator passes through a separator where any moisture is baffled off. It may then go to the distiller

and be condensed for use as drinking water, etc. If it is desired to use the condensed pure water for boiler make-up, it can be carried directly to the main engine condenser. The whole evaporator will then carry a vacuum: instead of "blowing" down, a pump must be used to withdraw concentrated brine: and the HEAT in the vapor (though not the vapor itself) will be discharged outboard (and therefore wasted) with the condenser circulating water. A better plan is to carry the vapor to the main hot well for condensation. Its heat as well as its substance is then transferred to the boiler feed.

Assume the distiller to be in use. Then with symbols as in Fig. 45,

Q_1 = heat supplied by live steam at evaporator

$$= w_8(h_8 + x_8L_8 - h_4),$$

Q_2 = heat absorbed at evaporator

$$= (w_5 + w_4)(h_3 + x_3L_3 - h_2) + w_6(h_6 - h_2),$$

$Q_1 = Q_2 +$ radiation at evaporator,

Q_3 = heat discharged at separator $= w_4h_3$,

$$w_4 = w_5\frac{1 - x_3}{x_3},$$

Q_4 = heat rejected at distiller $= w_5(H_3 - h_5)$,

Q_5 = heat absorbed at distiller $= (w_2 + w_4 + w_5 + w_6)(h_2 - h_1)$,

$Q_4 = Q_5 +$ radiation at distiller.

A test on a small machine gave the following values: $p_8 = 80$, $x_8 = 0.99$, $t_4 = 292.7$, $w_8 = 2800$, $p_3 = 25$, $x_3 = 0.99$, $t_2 = 126.15$, $w_5 = 2340$, $w_6 = 200$, $t_6 = 237.8$, $t_5 = 209.55$, $t_1 = 80$. The coil transmitting surface in the evaporator was 70 sq. ft. Weights are in lb. per hr. The following are tabular properties:

$h_8 = 282$, $L_8 = 900.3$, $h_4 = 262.1$, $h_3 = 208.4$, $L_3 = 952$,
$h_2 = 94$, $h_6 = 206.1$, $h_5 = 177.5$, $h_1 = 48.03$.

Then

$w_4 = 2340 \times \frac{1}{99} = 23.6$, $(w_5 + w_4) = 2363.6$,

$Q_1 = 2800\ (282 + \overline{0.99 \times 900.3} - 262.1) = 2\,552\,352$ B.t.u.,

$Q_2 = 2363.6\ (208.4 + \overline{0.99 \times 952} - 94) + 200(206.1 - 94)$

$= 2\,522\,420$ B.t.u.

Then the radiation loss at the evaporator is $Q_1 - Q_2 = 29\,932$ B.t.u.,

$Q_4 = 2340\ (208.4 + 952 - 177.5) = 2\,300\,000$ B.t.u.,

$Q_5 = (w_2 + 2363.6 + 200)(94 - 48.03)$

$= 45.97w_2 + 118\,000$ B.t.u.

Disregarding radiation at the distiller (it should be very small), put $Q_4 = Q_5$. Then $w_2 = 47\,400$ and the weight of circulating water pumped to the distiller is $47\,400 + 2563.6 = 49963.6$ lb., the greater part of which is wasted.

The heat transmitted at the evaporator was 2 522 420 B.t.u. per hr. The mean temperature difference (counter flow) was $t_m = 114$. The value of ρ realized was then $2\,522\,420 \div (70 \times 114) = 316$. Assuming the same value to have been realized at the distiller, with indeterminate flow and $t_m = 145$, the necessary surface there is $2\,300\,000 \div (145 \times 316) = 50$ sq. ft.

Higher values of ρ may be realized for a short period, but the accumulation of scale is rapid.

131. **Multiple Effect Evaporator.** In Art. 130, 2340 lb. of distilled water were produced per 2800 lb. of live steam condensed, or 0.833 lb. of water per lb. of steam. A large waste of heat occurred at the point w_2, h_2, Fig. 45. If the vapor leaving the separator had been conducted to the COIL of a second evaporator, in which the shell pressure was well below 25 lb., additional evaporation might have been produced there. Such an arrangement would constitute a MULTIPLE EFFECT machine.

Thus, in Art. 130, the vapor leaving the separator is at 25 lb. pressure and dry. Conduct to a second evaporator in which the shell pressure is 10 lb., the supply temperature 90° and the emerging vapor dry. Assume the coil drip to be at 230.6°. The weight of vapor delivered to this coil is $w_5 = 2340$ lb. The heat it delivers is $2340\ (H_{25} - h_{230.6}) = 2340\ (1160.4 - 198.8) = 2\,250\,000$ B.t.u. The heat required to vaporize 1 lb. of liquid fed to the shell is $H_{10} - h_{90} = 1143.1 - 58 = 1085.1$ B.t.u. The weight vaporized is then (ignoring radiation and blow down losses) $2\,250\,000 \div 1085.1 = 2070$ lb. In the two

effects combined, the distilled water produced per lb. of original steam condensed is $(2070 + 2340) \div 2800 = 1.57$. By using THREE effects, the economy might have been still further improved. Actual rates range up to 3.5 or 4 lb. of vapor per lb. of steam with SIX effects. The number of stages is limited by the total range of steam pressures. There must be a considerable temperature difference (and hence pressure difference) between coil and shell. The use of exhaust steam for the first or "primary" coil would make the complication and increased cost of multiple effect apparatus unnecessary: but with exhaust steam the mean temperature difference is low, the surface large and the apparatus expensive. Recent capital ships provide for the use of exhaust, but the amount of exhaust available is inadequate for continuous running. Weight limitations also preclude the exclusive use of exhaust steam.

When several stages are used, the latter ones are employed not for actual vaporization, but for heating the incoming salt water, just as the distiller heats it in Fig. 45. Matters are so adjusted that there is no surplus of water to waste between effects. As with heat transfer in general, parallel flow economizes surface while counter flow leads to a higher final temperature at the distiller discharge and hence permits of more stages (and more economy) with a given pressure range between primary steam supply and condenser. In parallel flow, the coldest salt water enters the first (highest pressure) stage: in counter flow, it enters the last (lowest pressure) stage.

Prob. 260. In Art. 130, check the values of t_m at evaporator and distiller.

Prob. 261. In Fig. 45, $w_8 = 2000$, $p_8 = 75$, $x_8 = 1.0$, $t_4 = 302°$, radiation at evaporator = 20 000 B.t.u. per hr., $t_2 = 153.01$, $p_3 = 30$, $x_3 = 1.0$, $w_6 = 0$. Find weight of distilled water produced per lb. of steam supplied. (Ans., 0.86 lb.)

Prob. 262. In Prob. 261, $t_1 = 90$, $t_5 = 205.87$. Radiation at distiller = 14 000 B.t.u. per hr. Find weight of salt water pumped through the distiller, and weight of this water that is wasted. (Ans., 26 900 lb. and 25 180 lb.)

Prob. 263. With $\rho = 300$, find surface necessary at evaporator and at distiller in Probs. 261 and 262. (Ans., 62 sq. ft. and 53 sq. ft.)

Prob. 264. In the foregoing, instead of using a distiller, the vapor from the evaporator is carried to a coil in a second evaporator. The pressure in the shell is here 20 lb. The coil drip is at 240.1°, the salt water supply at 141.52° and the vapor produced is dry. If radiation here is 18 000 B.t.u. per hr., and $\rho = 300$, find the necessary surface in this second effect. (Ans., 104 sq. ft.) How much distilled water is produced per lb. of original steam by the whole machine? (Ans., 1.635 lb.)

132. **Radiation Loss from Steam Pipes and Vessels.** For bare pipes or steam vessels with little or no air circulation and with only such steam circulation as is due to condensation, $\rho = 1.8$, as for steam radiators in the table of Art. 125. The value may rise to 5 or more if the steam temperature is very high and if there is a FLOW of steam. Commercial COVERINGS lead to values of ρ below 0.5.

Prob. 265. A 10-in. pipe ($10\frac{3}{4}$ in. outside diameter, 81.55 sq. in. internal cross-sectional area) carries steam initially dry at 200 lb. pressure, a distance of 300 ft. The steam velocity entering the pipe is 8000 ft. per min. The surrounding air is at 50° F. Take $\rho = 3$, the pipe being bare. How much heat is lost per hour? What weight of steam flows per hr.? What proportion of its original heat (above 32°) does the steam lose? Disregarding drop of pressure, what is the dryness of the steam at the far end of the pipe? (Ans., 841 000 B.t.u. lost per hr.: 118 400 lb. flow per hr.: loss = 0.59 per cent.: dryness = 0.99.) Note: the loss of heat goes on whether the pipe is delivering steam at full capacity or not: hence the PERCENTAGE of loss is great when the delivery of steam is small.

CHAPTER VIII

FLUID FLOW AND THE STEAM TURBINE

Flow of Steam

133. **New Assumption.** Thus far, Equation (3) has been our basis. There is one effect of heat not recognized in that Equation as heretofore applied, i.e., the production of KINETIC ENERGY. Where the addition of heat produces appreciable fluid velocities, the general equation must be written

$$H = T + I + W + V, \tag{73}$$

where V stands for that part of the heat H which is converted into kinetic energy or VELOCITY energy.

If 1 lb. of steam works from condition $_1$ to condition $_2$ without reception or emission of heat, the total energy being constant, and U representing velocity,

$$(T + I + W)_1 + \frac{U_1^2}{2g} = (T + I + W)_2 + \frac{U_2^2}{2g}.$$

Now $(T + I + W)$ may be taken as the total heat of steam above 32°, whether the steam be wet, dry or superheated. Designating such total heat by Q,

$$778Q_1 + \frac{U_1^2}{2g} = 778Q_2 + \frac{U_2^2}{2g}.$$

In most cases, U_1^2 is negligibly small* as compared with U_2^2. Hence, nearly,

$$U_2^2 = 778 \times 2g(Q_1 - Q_2),$$
$$U_2 = 224\sqrt{Q_1 - Q_2}, \tag{74}$$

where U_2 is the final velocity in ft. per sec. Values of Q_1 and Q_2 are to be taken as for adiabatic constant-entropy expansion

* In a steam turbine, U_1 is the velocity of steam at the entrance to the nozzle and U_2 is the velocity at the nozzle outlet. The ratio of U_2 to U_1 may be about 20: hence that of U_2^2 to U_1^2 is about 400.

from the total heat-entropy diagram, Fig. 26. The corresponding value of U_2 is the velocity attained by expansion in an ideal frictionless nozzle.

Prob. 266. Dry steam at 200 lb. pressure expands adiabatically to 1 lb. pressure. What is the ideal velocity attained? (Ans., 4100 ft. per sec.)

134. **Effect of Friction.** In all actual nozzles, there is a friction effect due to the rubbing of steam against metal. If there is sufficient friction, as when the orifice is very small, the velocity may be entirely destroyed. What then becomes of the heat developed by friction? It can only go BACK INTO THE STEAM. The law of such process (when the velocity is WHOLLY destroyed) is, $Q_1 = Q_2$: the total heat is constant. The expansion valve operation in vapor refrigeration, path *def*, Fig. 36, illustrates this. No heat has been absorbed or emitted from without: but the entropy has changed. Heat has been converted into velocity, and then reconverted, by means of friction, into heat.

In considering the injector (Prob. 240), nothing was said of velocity. The heat lost by the steam is first converted into velocity. There then follows an exchange of energies between water and steam, the kinetic energy is greatly reduced and the reduction is applied to increase the heat existing as such. In Prob. 240, it was assumed that the total heat was the same after mixture as before. This is not strictly true. A small part has become kinetic energy, represented by the moving body of warm water on its way to the boiler. At a boiler pressure of even 200 lb., each lb. of water must have velocity (kinetic energy) merely sufficient to overcome a head of about 460 ft., or the necessary kinetic energy is 460 ft. lb.—little more than half of 1 B.t.u. The error in Prob. 240 is therefore exceedingly small.

135. **Throttling Calorimeter.** This device consists essentially of a very small orifice through which steam is expanded. The velocity is wholly reconverted to heat, so that the law of the process is $Q_1 = Q_2$. Inspection of Fig. 26 will show that if the total heat remains constant while the pressure decreases, the

steam becomes drier and is finally superheated. The process is represented on Fig. 26 by a horizontal line drawn from left to right. The calorimeter is used to determine the quantity of moisture in steam. The pressure is measured on the inlet side, and pressure and temperature are noted on the outlet side. Then if the amount of expansion is sufficient to produce superheat at the outlet,

$$(h + xL)_1 = \{H + k(t' - t)\}_2. \tag{75}$$

Values of h_1, L_1, t_2 and H_2 are taken from the table, for the pressures noted. The observed value of t_2' then indicates the ORIGINAL dryness of the steam. The method fails when $t' \leqq t$, i.e., when no superheating occurs. This may result either from insufficient expansion or from the use of steam too wet to begin with. If a measured quantity of heat Q is added to each lb. of steam on the high-pressure side, the range of the instrument can be increased:

$$Q + (h + xL)_1 = \{H + k(t' - t)\}_2. \tag{76}$$

Problems may readily be solved by the use of Fig. 26, noting that a constant heat process implies a horizontal path.

Prob. 267. Steam at 100 lb. pressure is expanded to 15 lb. pressure in a throttling calorimeter. The temperature is then found to be 216°. What was the original dryness? (Ans. 0.961.)

Prob. 268. What is the wettest steam at 100 lb. pressure that can be tested in an ordinary throttling calorimeter using pressures as in Prob. 267? (Ans., 0.959 dry.)

Prob. 269. Steam at 100 lb. pressure has added to it 100 B.t.u. per lb. When expanded to 15 lb. its temperature is found to be 263°. What was its dryness? (Ans., 0.874.)

136. **Velocity and Flow in Actual Nozzles.** In Fig. 46, let a be the initial condition of the steam, determined by its pressure and quality. Let h indicate its condition at the same entropy but at some lower pressure, reached after expansion in a nozzle. Suppose friction to be such that 10 per cent. of the velocity

energy is re-converted to heat energy. Then lay off $hk = 0.10 \times ah = 0.10(Q_a - Q_h)$. The heat content after expansion is Q_k, but the pressure is p_h: hence the condition of the steam is indicated by the point j, found by drawing kj horizontally.

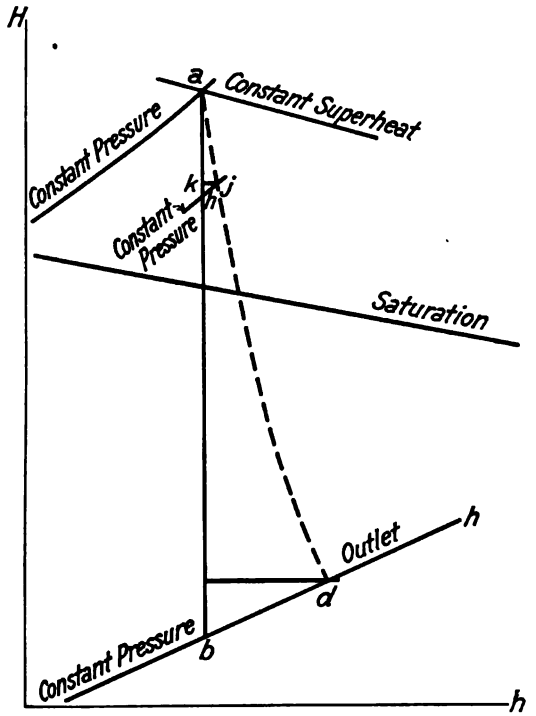

FIG. 46. Expansion in the Turbine.

Thus, if $p_a = 200$, superheat at $a = 200°$, and $p_h = 180$: Fig. 26 gives $Q_a = 1308$, $Q_h = 1297$. Then $hk = 0.10 \times 11 = 1.1$, $Q_k = Q_j = 1297 + 1.1 = 1298.1$. Fig. 26 now shows the steam to be 186° superheated at the point j. The velocity due to expansion from 200 lb. to 180 lb. is $224\sqrt{1308 - 1298.1} = 705$ ft. per sec. Note that the heat drop is $Q_a - Q_j$ instead of $Q_a - Q_h$, in an actual nozzle with friction.

Fig. 46*a* plots resulting velocities (U_2) for these conditions, with various pressures attained during expansion. The specific

volumes, v, of steam are also plotted on this diagram. Thus for an expanded pressure of 180 lb., with 186° of superheat, the specific volume is (Art. 83) 3.3 cu. ft.

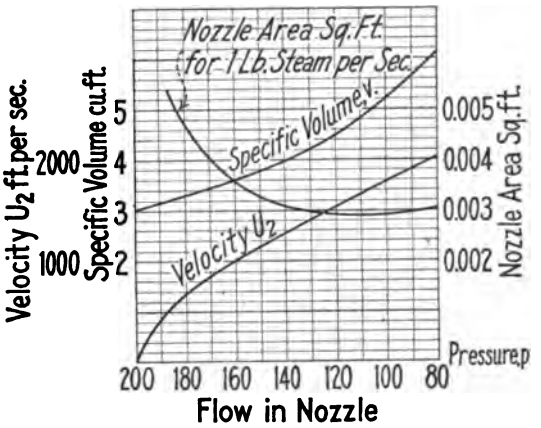

FIG. 46*a*. Flow in Nozzle.

If w_0 = weight of steam discharged, lb. per sec., the cross-sectional area of the nozzle, at a point where the pressure is p, is $vw_0 \div U_2$ sq. ft. Values of v/U_2 are plotted in Fig. 46*a*: which is based on an initial pressure of 200 lb., 200° initial superheat and 10 per cent. reconversion of kinetic energy to heat.

Suppose the "nozzle" is merely a hole in a plate. In this as in any other case, the area of cross-section is mathematically infinite on the high-pressure side, for here $U_2 = 0$. In practice, this side is simply well chamfered. If the pressure of discharge is 180 lb., the area for 1 lb. discharge per sec. is (from Fig. 46*a*) 0.0047 sq. ft. If the discharge pressure is 140 lb., the area per lb. discharged per sec. is 0.0031 sq. ft. Suppose the actual area to be 1 sq. ft. Then the weights discharged per sec. are $1 \div 0.0047 = 212$ lb. if the discharge pressure is 180 lb., and $1 \div 0.0031 = 323$ lb. if it is 140 lb. If the discharge pressure is 116 lb., the weight is a MAXIMUM, and is $1 \div 0.00291 = 344$ lb. per sec. The diagram shows clearly that the area necessary to discharge a given weight is a MINIMUM when the discharge pressure

is 116 lb. Conversely, the weight discharged through an orifice of given area is a MAXIMUM when the discharge pressure is 116 lb. A further decrease in discharge pressure does not increase the weight flowing. The discharge pressure at which maximum weight of flow is realized is called the CRITICAL PRESSURE. For steam, the critical pressure is always 58 per cent. of the initial pressure (it is 116 lb., in this case). For AIR, it is 53 per cent.

Now consider an actual nozzle designed for 200 lb. initial pressure and 200° initial superheat, with 10 per cent. reconversion. It will have a well rounded entrance. If it is a CONVERGING nozzle, it will decrease in cross-sectional area from inlet to outlet. Suppose, in such case, the outlet area to be 1 sq. ft. The weight of flow will depend on the outlet pressure and will be (as above) 212 lb. if the discharge pressure is 180 lb., 323 lb. if the pressure is 140 lb., and 344 lb. if it is the critical pressure of 116 lb. The corresponding outlet velocities, also from Fig. 46*a*, are 705, 1290 and 1590 ft. per sec. If the discharge pressure IS the critical pressure (116 lb.) then the pressure will be 140 lb. at a point in the nozzle where the cross-sectional area is 0.0031/0.00291 = 1.065 sq. ft., and the velocity at that point will be 1290 ft. per sec.

If the discharge pressure is LESS than the critical pressure, the nozzle should first contract to a THROAT and then DIVERGE (diverging nozzle). Thus in Fig. 46*a*, the pressure in the throat (narrowest section) is 116 lb. The ordinate of the nozzle area curve is there 0.00291. At a point beyond the throat, where the cross-sectional area is 1.047 sq. ft., the ordinate corresponding is $1.047 \times 0.00291 = 0.00305$, so that the pressure at that point is 80 lb. and the velocity is 2030 ft. per sec. Note that THE DIVERGENT PORTION OF THE NOZZLE DOES NOT AFFECT THE WEIGHT OF FLOW, BUT ONLY THE FINAL (OUTLET) PRESSURE AND VELOCITY. The weight of flow is determined by the throat area, and the throat pressure is the critical pressure; IF the final pressure is less than the critical pressure.

The reason for these relations is suggested by Fig. 46*a*. As expansion proceeds, pressure falls and volume and velocity both

increase. The volume at first increases more slowly than the velocity, and afterward more rapidly. The stream of steam reaches its minimum cross-section where the quotient of volume by velocity is a minimum.

The long equation of Art. 83 need be used for specific volumes ONLY WHEN THE STEAM IS SUPERHEATED. After the expansion passes the saturation curve (Fig. 46), the dryness may be read from Fig. 26 and the corresponding wet steam volume computed directly.

Approximate formulae for flow in lb. per HOUR are

$$\left.\begin{aligned} w &= \frac{60A_t p_a^{0.97}}{\sqrt{x_a}} \text{ for saturated steam} \\ w &= \frac{60A_t p_a^{0.97}}{1 + 0.00065(t_a - t)} \text{ for superheated steam} \end{aligned}\right\} \quad (77)$$

where A_t = throat area in sq. in., $(t_a - t)$ = superheat. These apply only when the discharge pressure is less than 0.58 the initial, and are based on a reconversion of about 7 per cent. instead of 10 per cent. For AIR, when the discharge pressure is equal to or less than 0.53 the initial,

$$w = \frac{1900 p_a A_t}{\sqrt{T_a}}. \quad (78)$$

The velocity in the throat is then that of SOUND.

Prob. 270. (*a*) Steam at 200 lb. pressure, 200° superheated, is discharged through a converging nozzle with 10 per cent. reconversion to a chamber where the pressure is 40 lb. The outlet area is 0.1 sq. ft. What is the pressure in the outlet? (Ans., 116 lb.) What weight of steam will be discharged? (Ans., 34.4 lb. per sec.) What is the outlet velocity? (Ans., 1590 ft. per sec.) (*b*) The discharge pressure is changed to 140 lb. State outlet pressure, velocity and weight discharged. (Ans., 140 lb., 1290 ft. per sec., 32.3 lb.) (*c*) A DIVERGING nozzle having the same reconversion is used for the same initial steam conditions. It discharges 2000 lb. per hr. with a discharge pressure of 1 lb. Find pressure, velocity, specific volume and

area at the throat and at the outlet. (Ans., at THROAT, 116 lb., 1590 ft. per sec., 4.6 cu. ft., 0.00161 sq. ft. At OUTLET, 1. lb., 4130 ft. per sec., 290 cu. ft., 0.0389 sq. ft.) (*d*) Under case (*c*), state values of TOTAL VOLUME PER HR. ÷ VELOCITY IN FT. PER SEC. at throat (Ans., 5.79), outlet (Ans., 140), and at a point in the expansion where the heat content is 1274.7 and the specific volume is 4 (Ans., 6.2).

Prob. 271. Check Prob. 270, Case (*c*), as to weight flowing, by Equation (77). (Ans., Equation gives 2100 lb. per hr.) What weight of DRY steam at 200 lb. pressure would flow through the same nozzle? (Ans., 2400 lb.) What weight of air at this pressure and at 40° F.? (Ans., 3900 lb.) What would be the throat pressure with air? (Ans., 106 lb.) What is the throat velocity with air? (Ans., 1180 ft. per sec.)

Prob. 272. Determine throat and outlet areas of a nozzle to discharge 500 lb. of dry steam at 200 lb. pressure, per hr., if the discharge pressure is 2 lb. and the reconversion is 10 per cent. (Ans., 0.0528 and 0.768 sq. in.)

137. **Nozzle Efficiency.** The percentage of reconversion is determined by the design of the nozzle. If there were no friction, there would be no reconversion and the nozzle efficiency would be 100 per cent. Good efficiency requires smooth interior finish, a well rounded entrance and a proper angle (15° or so) between the diverging sides. Most important, the outlet area and throat area should be in suitable ratio for the pressures used. If either pressure (inlet or outlet) differs from that contemplated in design, eddies will be produced and a loss of efficiency will result. If e = nozzle efficiency, $1 - e$ = proportion of reconversion and $U_2 = 224\sqrt{e(Q_a - Q_b)}$, Fig. 46. The value of U_2 varies, thus, not with e but with $\sqrt{e}$. Values of e up to 0.94 may be expected. Note that $e = U_2^2 \div 50\,056\ (Q_a - Q_b)$.

Prob. 273. The nozzle efficiency is 0.94. How much is velocity reduced by friction? (Ans., 3 per cent.)

Simple Impulse Turbines

138. **Simple Velocity Diagram.** The small sketch in Fig. 46*b* indicates the operation of a single-stage impulse turbine, like the DE LAVAL. The wheel *W* carries curved blades or BUCKETS, *B* and revolves with its shaft *S*. The nozzle *N* directs the jet at an acute angle (about 20°) with a plane of rotation.

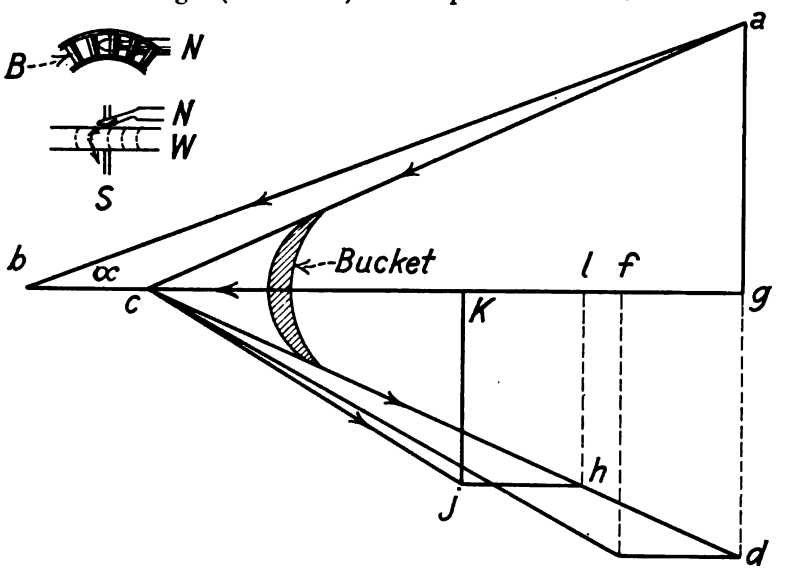

FIG. 46*b*. Impulse Turbine; Single Expansion.

The larger diagram shows the velocities involved. This diagram lies in an imaginary plane which, in the upper of the sketch views, passes through the axis of the nozzle and is perpendicular to the plane of the paper.

Draw *gb* representing the line of motion of the bucket. Draw *ab* representing the direction of motion of the steam jet, the two lines intersecting at the NOZZLE ANGLE α. Let the lengths *ab* and *bc* represent the magnitudes of steam and (peripheral) wheel velocities. Now the RELATIVE* steam velocity (velocity

* ABSOLUTE AND RELATIVE VELOCITIES. (1) A ship sails west at 10 knots speed. The wind blows west at 14.17 knots (absolute velocity of wind). The RELATIVE velocity of the wind WITH RESPECT TO THE SHIP is 4.17 knots,

as it would seem to be to an observer moving with the wheel) is the resultant *ac*. This is deflected by the bucket to the direction *cd*. Friction and eddy losses make the ultimate relative

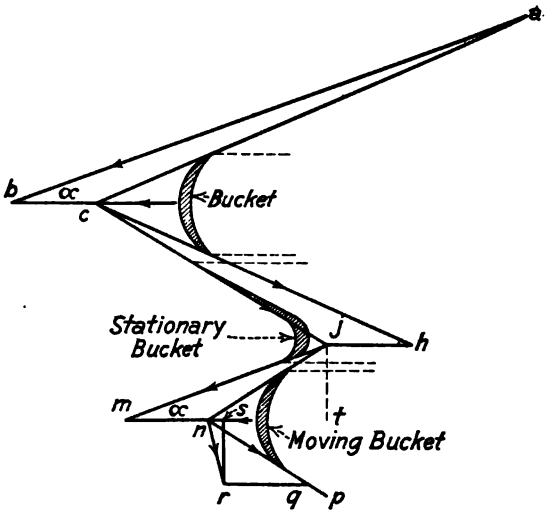

FIG. 47. Impulse Turbine Triple Expansion.

exit velocity less than the initial relative velocity, the relation of the two being as in Fig. 48. Let *ch* denote the magnitude of the relative exit velocity. Combine this with the wheel velocity ($hj = bc$) to obtain the absolute or true velocity, *cj*, of steam leaving the buckets.

Force is change of momentum. For 1 lb. of steam, the initial momentum is ab/g. The momentum IN THE DIRECTION OF MOVEMENT is bg/g. The final momentum in the same direction is $-ck/g$. The negative sign indicates that this is a REARWARD momentum. The corresponding force is a reactive impulse, or

westward. (2) The wind blows EAST at 14.17 knots. The absolute wind velocity is 14.17 knots, the relative velocity is 24.17 knots, east. (3) The wind blows SOUTH at 10 knots absolute velocity. The RELATIVE velocity will be 14.17 knots, SOUTHEAST. (4) The wind blows SOUTHWEST at 14.17 knots absolute velocity. The relative velocity is 10 knots, SOUTH. In each case, the relative velocity is directed aft as compared with the absolute velocity.

REACTION, while the force corresponding with the momentum bg/g is an IMPULSE. The change of momentum, or force impelling the bucket, is $(bg + ck)/g$. The speed of the bucket is bc.

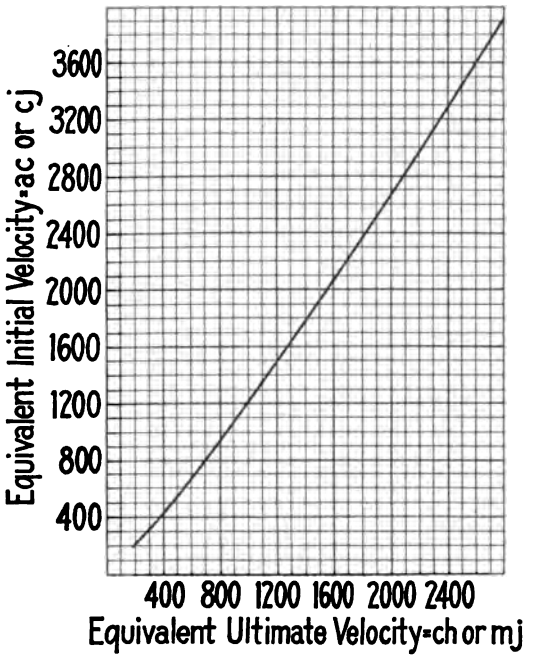

FIG. 48. Bucket Friction.

Work is force times speed. Hence the work obtained per lb. of steam is

$$W = \frac{bc}{g}(bg + ck) \text{ ft. lb.}$$

$$= \frac{bc(bg + ck)}{32.17 \times 778} = \frac{bc(bg + ck)}{25028} \text{ B.t.u.} \quad (79)$$

Prob. 274. Steam at 150 lb. pressure, and dry, expands to 1 lb. pressure with a nozzle efficiency of 0.93. Find the resulting velocity. (Ans., 3850 ft. per sec.)

Prob. 275. With steam velocity = 3850, bucket velocity = 600, $\alpha = 20°$, construct to some convenient scale a velocity

diagram like that of Fig. 46*b*, the relative angles *acg* and *dcg* being equal. Measure the necessary lines and state the work done per lb. of steam. (Ans., 125 B.t.u.)

139. **Kinetic Efficiency.** The efficiency of conversion of kinetic energy into work AT THE BUCKETS is what we may call the KINETIC EFFICIENCY,

$$e_k = W \div \frac{U_{ab}^2}{2g \times 778} = \frac{50\,056W}{U_{ab}^2}. \tag{79}$$

It is easily shown that the maximum value with equal relative angles is attained when $bc = \frac{1}{2}bg$, and that that value is

$$e_{k\,\max} = \left(\frac{ab}{bg}\right)^2 \times \frac{1+F}{2},$$

where F = bucket friction coefficient = $ch/ca = ch/cd$, to be taken from Fig. 48. If there is no bucket friction, $F = 1$ and $e_{k\,\max} = (bg/ab)^2$. If $\alpha = 0$, then the point k would coincide with the point f and $e_{k\,\max}$ would be 100 per cent. when $bc = \frac{1}{2}ba$. Note that α is a necessary evil and that the ideal performance is realized when cj is vertical.

MAXIMUM KINETIC EFFICIENCY OCCURS WHEN THE PERIPHERAL SPEED OF THE BUCKETS IS HALF THE ROTATIVE COMPONENT OF THE STEAM SPEED. High peripheral speeds are essential to efficiency.

Prob. 276. Compute e_k in Prob. 275. (Ans., 0.42.)

Prob. 277. With $\alpha = 0°$, Fig. 47, $gcd = 0°$, $F = 1.0$, what would have been the best value of bc, W and e_k for Prob. 275? (Ans., $bc = 1925$ ft. per sec., $W = 297$ B.t.u., $e_k = 1.0$.)

Prob. 278. In Prob. 275, for $\alpha = 20°$, $F = 1.0$, state the best value of bc and the resulting values of W and e_k, the relative angles being equal. (Ans., $bc = 1810$ ft. per sec., $W = 261$ B.t.u., $e_k = 0.88$.)

Prob. 279. With the bucket friction and value of α assumed in Prob. 275, and with equal relative angles, state the best values of bc, W and e_k. (Ans., $bc = 1810$, $e_k = 0.76$, $W = 226$ B.t.u.)

Fig. 49 shows the variation in W with bucket speed, bc, for the conditions of Prob. 275, except that the friction factor F is taken constantly at 0.75. The variation of F is such that the work is actually a little greater than is indicated by the curve, at high speeds, and a little lower at low speeds. Note that the curve of Fig. 49 represents e_k (to another scale).

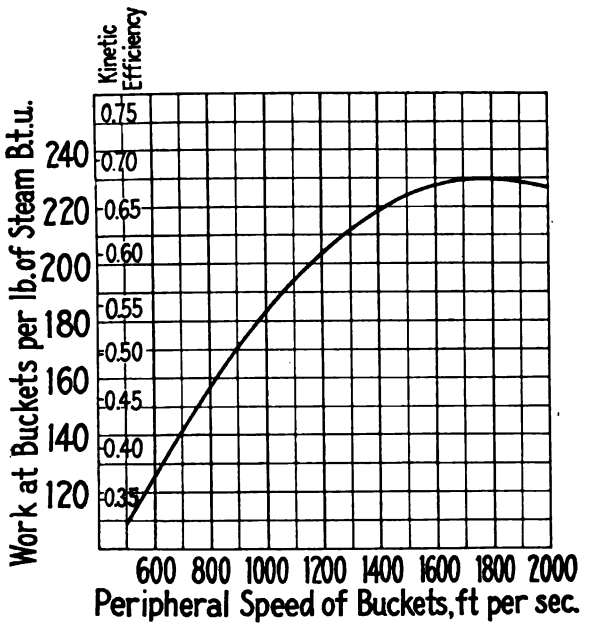

FIG. 49. Effect of Bucket Speed on Work at Buckets and on Kinetic Efficiency.

140. **Primary Difficulties in Turbine Design.** Let u (= bc, Fig. 46b) denote peripheral speed in ft. per sec., N = r.p.m., d = wheel diameter (pitch diameter) in ft. Then $\pi dN = 60u$ or $dN = 19.1u$. For high values of u, either d or N must be high. The smaller the capacity, the smaller preferably, is d: hence N is high in small machines. Flexible shafts and reduction gears may be essential in turbines of this type. The largest values of u in actual use are around 1300; of N, around 24 000.

The capacity of an impulse turbine is determined by the nozzle

area and steam conditions rather than by any dimension of the wheel. The height of the buckets should somewhat exceed the nozzle outlet diameter, to prevent lateral spreading of steam. The wheel runs in a chamber containing steam at condenser pressure, and (in this type) need not make a close fit with its casing. No high pressure steam is carried beyond the nozzle. The pressure is constant across the face of the buckets.

Prob. 280. In Prob. 275, what is the pitch diameter of the bucket wheel if it makes 10 000 r.p.m.? (Ans., 1.146 ft.)

141. **Turbine Efficiency.** The turbine must be regarded as operating in the complete expansion (Rankine) cycle: Art. 80. Its ideal efficiency is therefore to be computed as in Equations (45), (49). We have seen that the conversion of heat into kinetic energy is accompanied with a nozzle friction loss, and that the conversion from kinetic energy into work at the buckets is accompanied with a further loss. Let e_I = efficiency of ideal cycle, e_N = nozzle efficiency, e_k = kinetic efficiency. Then the efficiency from steam to work at the buckets is $e_I \times e_N \times e_k$. Further loss now occurs.

The revolving disk and the rows of buckets both undergo friction against the steam. For wheels under 36 in. diameter revolving in a chamber containing dry steam, empirically,

$$\left.\begin{array}{ll} L_d = d^2u^{2.9} \div 4400v & \text{B.t.u. per hr.,} \\ L_b = dh^{1.25}u^{2.9} \div 990v & \text{B.t.u. per hr.,} \end{array}\right\} \qquad (80)$$

where v = specific volume of dry steam at nozzle outlet pressure, h = radial height of buckets, in., L_d = friction loss at disk, L_b = friction loss for one row of buckets. For m rows of buckets on one disk, the total ROTATION LOSS per disk is $L_d + mL_b$. This is to be multiplied by the factors of Fig. 50 if the steam is (as usually) not dry in the wheel chamber. In Fig. 46, the steam enters the chamber as it leaves the nozzle, at the state d. It leaves the chamber at a state h, defined by $Q_a - Q_h = W$, the pressure at h being the same as at d. The MEAN of the drynesses at d and h determines the abscissa of Fig. 50.

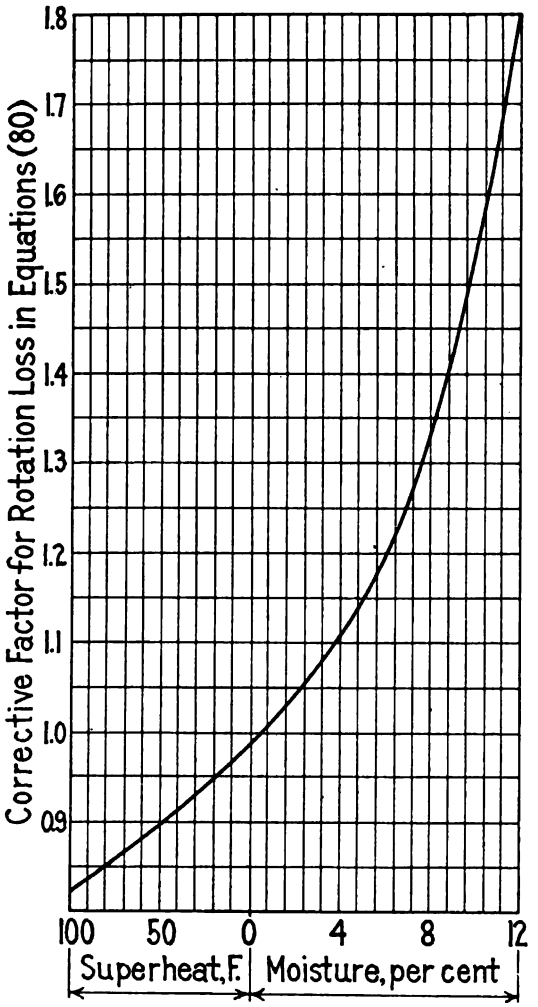

FIG. 50. Correction Factors for Rotation Loss.

It should be noted that the amount of rotation loss depends on the dimensions of the wheel and not on the weight of steam flowing. Hence it is relatively large for machines of small output or with large diameter wheels. If two rows of buckets are placed on one disk, L_d is not increased. Hence the total rota-

tion loss is not doubled. Rotation loss is reduced when the steam is initially superheated. It is very great where buckets revolve in very wet steam. For wheels larger than 36 in., L_d may be 5 per cent. greater than is indicated by Equation (80).

Prob. 281. A wheel of 1.146 ft. dia. revolves at 10 000 r.p.m. in dry steam at 1 lb. pressure. Buckets are 1.3 in. high. Find L_d and L_b. (Ans., 103 and 560 B.t.u. per hr.)

Prob. 282. The wheel in Prob. 281 revolves in a chamber in which the pressure is 1 lb. The nozzle inlet pressure was 150 lb. (dry steam) and the nozzle efficiency 0.93. What is the dryness at the nozzle outlet? (Ans., 0.80.) If the work at the buckets is 125 B.t.u., find Q_h. (Ans., 1069.) What then is x_h? (Ans., 0.966.) What is the mean dryness in the wheel chamber? (Ans., 0.883.) What is the rotation loss correction? (Ans., 1.765.) What is the corrected rotation loss? (Ans., 1170 B.t.u. per hr.)

142. **Efficiency and Steam Rate.** The bearing friction and shaft packing losses of the turbine may be kept down to 1 per cent. of the shaft output. Radiation losses are small. Losses incurred in speed reduction are perhaps properly chargeable to the turbine in comparing its performance with that of a reciprocating engine. Efficiencies of REDUCTION GEARS range up to 0.98: hydraulic transmissions and electric drive are technically less efficient. The WORK AT THE BUCKETS corresponds with the Ihp. of an engine, but cannot be measured directly. Turbine capacities and economics are therefore expressed on the basis of brake hp. or shaft hp. The ratio of shaft hp. to bucket hp. may be called the MECHANICAL EFFICIENCY, although the meaning of this term is then not quite the same as with steam engines. This usage lumps the rotation loss with the mechanical friction loss.

Let W = work at buckets per lb. of steam, in B.t.u., w = lb. of steam per hr. Then bucket hp. = $Ww \div 2545$. Let L = total rotation loss, B.t.u. per hr. Then shaft input = $Ww - L$ B.t.u. per hr. Let $1 - l$ = proportion of shaft input lost in

mechanical friction (say 1 per cent.). Then $l(Ww - L)$ = shaft output, B.t.u. per hr., or $l(Ww - L)/2545$ = shaft hp. and the mechanical efficiency, as above defined, is

$$l(Ww - L) \div Ww = \frac{2545 \times \text{shaft hp.}}{Ww} = e_M.$$

The efficiency from steam to work at the shaft is $e = e_I \times e_N \times e_k \times e_M$. What corresponds with the "relative efficiency" of the steam engine (Art. 93) is the product $e_k \times e_N$. This may be higher than in steam engines, and is chiefly determined by the value of e_k. If (Fig. 46) the steam and feed water conditions are Q_a and h_b, the weight of steam per brake hp.-hr. is as in Art. 93, $w/\text{hp.} = 2545 \div e(Q_a - h_b)$ lb. Fig. 51 gives ideal steam rates for the complete expansion cycle (Arts. 80, 84). Results from actual trials are given below, Art. 148.

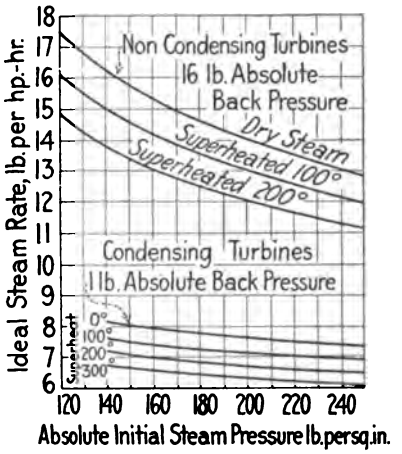

FIG. 51. Steam Rates for Ideal Turbine Cycles.

Prob. 283. The turbine gives 125 B.t.u. work at buckets per lb. of steam, and the rotation loss is 1170 B.t.u. per hr. Mechanical friction is 1 per cent. The shaft hp. is 100. Find weight of steam per hr. (Ans., 2053 lb.)

Prob. 284. In Prob. 283, what is the "mechanical efficiency"? (Ans., 0.985.) If the initial steam was at 150 lb. pressure, and dry, and the back pressure was 1 lb., with $e_N = 0.93$, find e_I, e_k, and e. (Ans., 0.28, 0.42, 0.108.) What is the "relative efficiency"? (Ans., 0.39.) What weight of steam is used per brake hp.-hr.? (Ans., 20.53 lb.) Check the last result from Fig. 51.

Multi-Stage Turbines

143. **Velocity Compounding.** In Fig. 46*b*, there is a considerable exit velocity, *cj*, which carries away with it kinetic energy. This velocity should be kept small, or else means should be provided for the further utilization of the wasted kinetic energy. The method is suggested in Fig. 52. The steam leaving the buckets *A* is reversed by stationary buckets *B*, projecting inward from the casing, and USED AGAIN in a second set of buckets *C*. To decrease rotation loss (Art. 141), these additional buckets are mounted on the original disk.

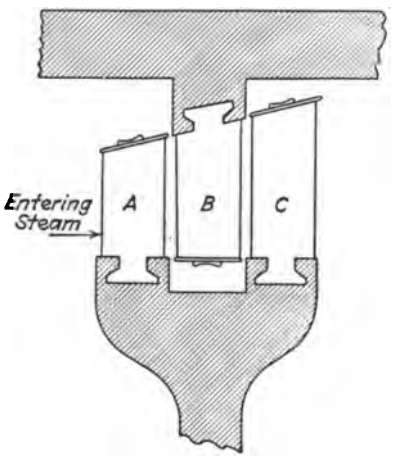

FIG. 52. Arrangement of Buckets in a Velocity-Compounded Turbine.

In Fig. 47, the triangles *abc*, *chj* duplicate those of Fig. 46*b*. Now lay off *jm* parallel with *ba*, implying equality of angles α for both rows of buckets. The length of *jm* is that of *cj*, reduced by bucket friction as in Fig. 48. Lay off *mn*, parallel and equal to *bc*. (Both rows of buckets move in the same direction at the same speed.) Draw *jn* and draw $np = jn$ so that the relative angles *jnt* and *pnt* are equal. (This equality does not always exist.) Lay off *nq*, less than *np*, the ratio of the two being given by Fig. 48. Lay off *qr* parallel and equal to *mn*. Then the ADDITIONAL WORK gained by the second row of moving buckets, AT NO ADDITIONAL COST for heat, is

$$W_2 = \frac{mn(mt + ns)}{25\,028} \text{ B.t.u. per lb. of steam.} \qquad (81)$$

If the point *r* lies to the LEFT of a vertical line through *n*, the component *ns* is NEGATIVE.

Such velocity compounding increases work and efficiency. In the present illustration, only two stages are used. This is, in fact, all that it would be profitable to use under the velocity relations shown by Fig. 47 to be existing. With lower values of u, the desirable number of stages increases. By velocity compounding, we may increase the kinetic efficiency at a given value of u, or may maintain some desired efficiency without using high values of u. Bucket friction establishes a limit to this procedure.

Prob. 285. In Prob. 275, a second stage is added as described in Art. 143. What is the additional work obtained per lb. of steam? (Ans., 35 B.t.u., or 28 per cent.)

144. **Pressure-Compounded Impulse Turbine (Curtis Type).** It is the RATIO of bucket speed to steam speed which determines

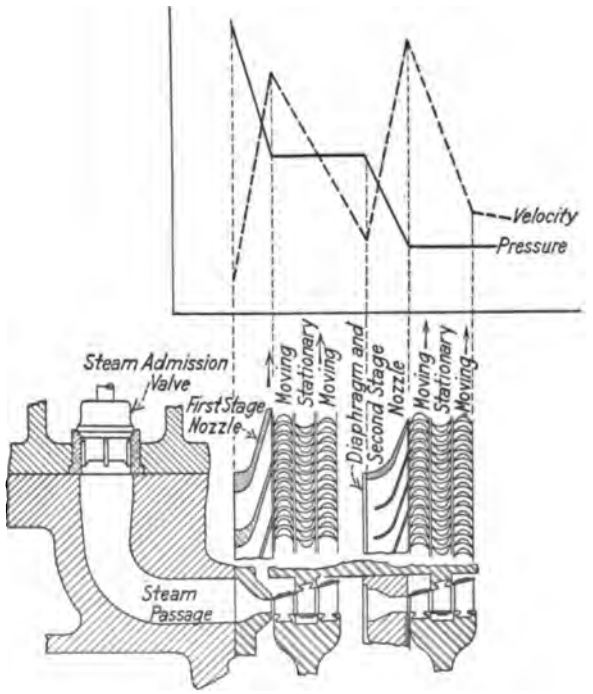

FIG. 53. Arrangement of Nozzles and Buckets in a Two-Stage Curtis Turbine.

kinetic efficiency, rather than the bucket speed itself. Velocity compounding, as just described, is equivalent to increase of bucket speed but is limited by bucket friction. PRESSURE COMPOUNDING attacks the problem from another angle, by decreasing the steam speed. This is done by expanding in stages. One set of nozzles expands the steam PART WAY down to condenser pressure. If HALF the "heat drop" *ab*, Fig. 46,

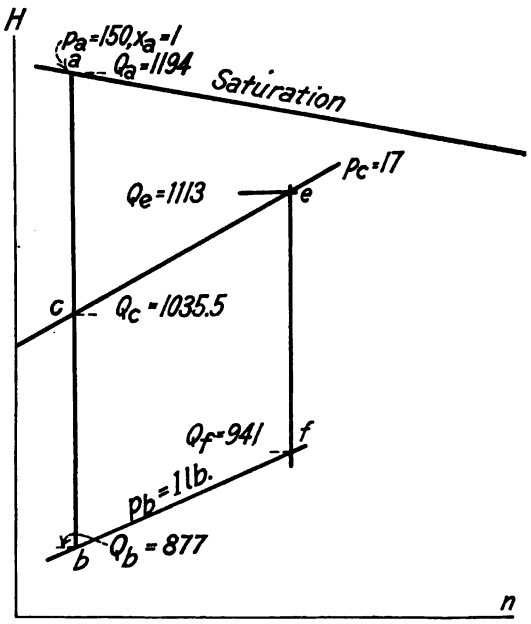

FIG. 54. Pressure-Compounded Turbine.

is realized in this stage (or set of nozzles), the outlet velocity resulting will be that in a single-expansion nozzle divided by $\sqrt{2}$ (Equation (74)). This velocity is then used in a wheel which may be either simple or velocity-compounded. Leaving this wheel, the steam expands down to condenser pressure through the second set of nozzles and then strikes a second wheel. Fig. 53 suggests the arrangement with two PRESSURE STAGES (sets of nozzles) and two-stage velocity compounding in each pressure

stage. The number of pressure stages should increase as the peripheral speed, u, decreases, in order to maintain an advantageous ratio between u and the steam velocity.

Assume steam at 150 lb. pressure, dry, with a condenser pressure of 1 lb. and a nozzle efficiency of 0.93. Locate the initial state, a, on the total heat-entropy diagram, as in Fig. 54. Draw ab vertically, locating b. Read off $Q_a = 1194$, $Q_b = 877$.

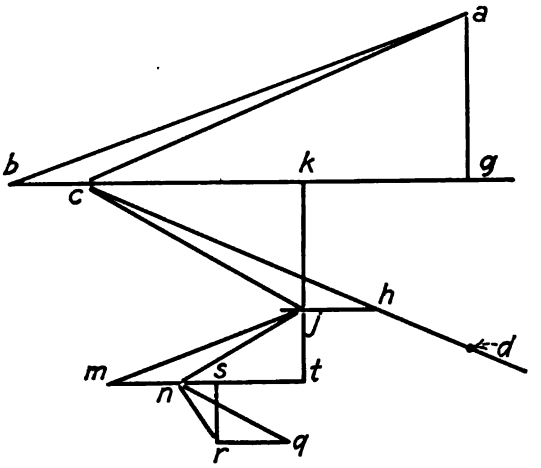

FIG. 55. Two-Stage Velocity Diagram, Pressure-Compounded Turbine.

Assume two pressure stages: therefore bisect ab, determining $p_c = 17$ lb., $Q_c = 1035.5$. The first stage nozzles will expand from 150 to 17 lb. pressure and the velocity resulting is $224\sqrt{0.93(1194 - 1035.5)} = 2720$. Assume $u = 400$, $k = 20°$ and equal relative angles. Construct the velocity diagram for two velocity stages as in Fig. 55, following Fig. 47. The work per lb. of steam in the first pressure stage is

$$\frac{bc}{25\,028}(bg + ck + mt + ns)$$

$$= \frac{400}{25\,028}(2550 + 1230 + 1095 + 215) = 81 \text{ B.t.u.}$$

The heat in the steam entering the second pressure stage is (Fig. 54) $Q_e = Q_a - 81 = 1113$. This determines the point e on

the line of 17 lb. pressure. Adiabatic expansion in the second pressure stage is represented by the vertical line *ef*, determining $Q_f = 941$ B.t.u. Then the heat to be converted into velocity in this second pressure stage is $Q_e - Q_f = 1113 - 941 = 172$ B.t.u. In the first stage it was $1194 - 1035.5 = 158.5$ B.t.u. The second stage will therefore have a higher initial velocity and will give more work per lb. of steam.

It is usually desirable that the stage conversions be equal: i.e., that $Q_a - Q_c = Q_e - Q_f$. This leads to similar blading in the two stages. The conversions are UNEQUAL in this case because of REHEAT, i.e., the reconversion of velocity not utilized, into heat. To allow for the influence of reheat, we may arbitrarily increase the drop in the first stage. This implies lowering the discharge pressure of the nozzles in that stage. An approximation to the desired condition is obtained by starting with a heat drop which is a mean between $Q_a - Q_c$ and $Q_e - Q_f$.

Prob. 286. What is the mean value of the two heat drops in Art. 144? (Ans., 165.3.) If this drop is used for the first stage, what is the value of Q_c? (Ans., 1028.7.) What is then the

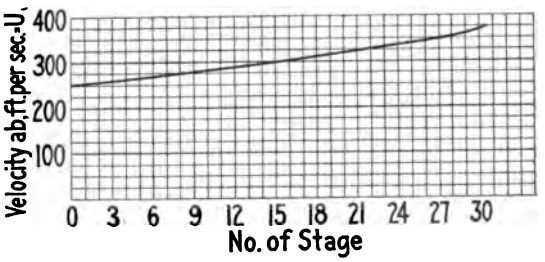

FIG. 56. Steam Velocity, Reaction Turbine.

value of p_c? (15.7.) Of the outlet velocity? (Ans., 2780.) Assume that the use of this velocity in the first stage velocity diagram, Fig. 55, gives work equal to 84 B.t.u. per lb. of steam. What then is the value of Q_e? (Ans., 1110 B.t.u.) Of p_e? (Ans., 15.7.) Of Q_f? (Ans., 944.7 B.t.u.) Of the heat drop in the second stage? (Ans., 165.3.) Of the work in the second stage? (Ans., 84 B.t.u.) What is the total work at the buckets

of the whole turbine? (Ans., 168 B.t.u. per lb. of steam.) What is the gain over the machine of Prob. 275? (Ans., 43 B.t.u. or 34 per cent.) Note that this gain is secured even though u has been decreased.

145. **Reaction Turbine.** The impact due to the velocity cj, Fig. 46*b*, is a REACTIVE impact. The steam is flowing AWAY from the bucket. This reaction is less than the impulse produced by the inlet velocity ab. Under ordinary conditions, it MUST be less. If, however, the buckets are so shaped that some expansion (lowering of pressure) occurs in them, there may be an increase of velocity instead of a decrease, from ac to ch, in spite of bucket friction. The reaction work may then equal or even exceed the impulse work.

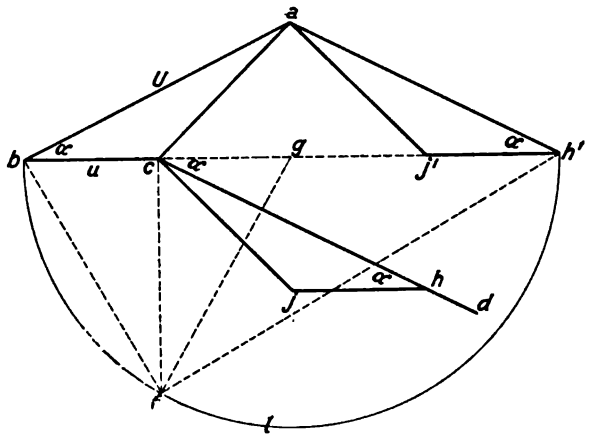

FIG. 57. Velocity Diagram, Reaction Turbine.

In Fig. 57, assume that the RELATIVE EXIT angle gcd is equal to the ABSOLUTE INLET angle, α. Assume also that there is expansion in the buckets, so great that the relative exit velocity without friction would be $cd > ab$. Assume further that the amount of expansion is such that this velocity, reduced by friction as in Fig. 48, is $ch = ab$. Transpose the triangle chj to the position $ah'j'$. Here it will be symmetrically opposed to the triangle abc.

The work per lb. of steam, following Equation (79), is

$$\frac{bc}{25\,028}(bg + gc) = \frac{bc \times ch'}{25\,028}\text{ B.t.u.}$$

About g as a center describe a circular arc bl. Draw cf vertically. Then, from geometry, $bc \times ch' = \overline{cf}^2$ and

$$W = \frac{\overline{cf}^2}{25\,028} = \left(\frac{cf}{158.5}\right)^2 \text{B.t.u.} \qquad (82)$$

This modified system of velocities is approximated in turbines of the REACTION (Parsons) type. There is a further characteristic of reaction turbines. Since expansion is to occur in the blades, NO NOZZLES ARE NECESSARY. The steam entering the first row of blades is simply at steam pipe velocity, around 100 to 200 ft. per sec. Hence peripheral speeds (which should be about half the steam speed) may be very low without sacrifice of efficiency. Each row of blades may be regarded as a set of nozzles designed for a very small heat drop. The reaction turbine is essentially a low speed turbine. It contains rows of moving buckets alternated with rows of stationary reversing buckets (like those of Fig. 47), projecting inward from the casing.

Prob. 287. In Fig. 57, $ab = 224$, $bc = 100$, $\tan \alpha = 0.5$. Find W. (Ans., 1.2 B.t.u.)

146. **Steam Velocities in Reaction Turbines.** With low steam velocities, the work done in a row of buckets is necessarily very small. This is suggested by Prob. 287. Hence the number of rows is very large. If the average work per row is 5 B.t.u., and the total bucket work of the turbine 150 B.t.u., 30 stages or rows of buckets are required.

Fig. 57 shows that the absolute exit velocity cj is less than the absolute entrance velocity ab. In passing to the second row of moving buckets, however, further expansion occurs in the fixed blades: so that the absolute entrance velocity of the second row is made somewhat GREATER than ab. Hence (if the value of u is the same for both rows) a greater amount of work will be done in the second row than in the first.

A reaction turbine, unlike one of the impulse type, carries its wheel chamber constantly full of steam. Its capacity is therefore determined by the area of the annulus comprising the buckets, as well as by the steam conditions. Tip clearances must be low. This is not necessary in impulse turbines.

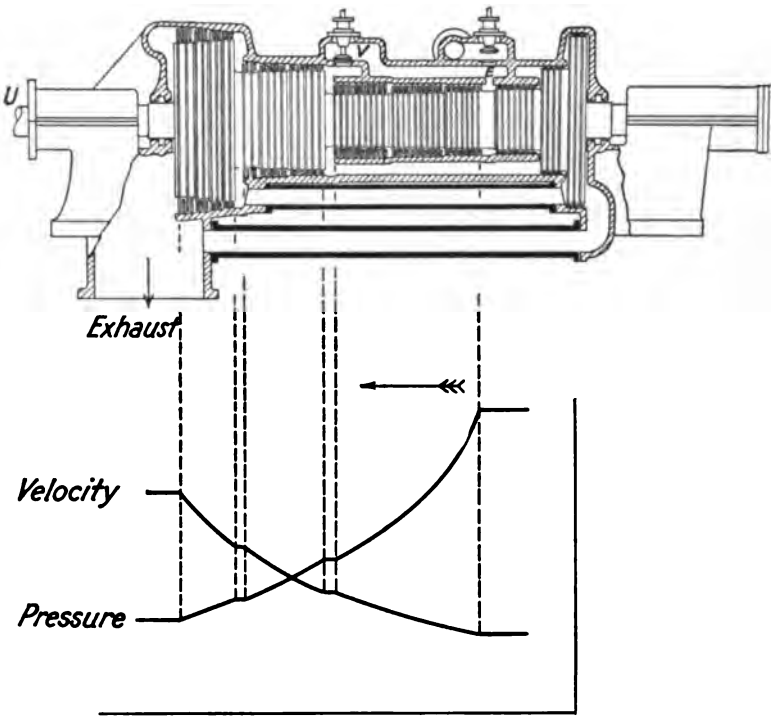

FIG. 58. Parsons Reaction Type Steam Turbine.

Fig. 58 shows a marine type of reaction turbine. Steam enters at E and moves to the left. The buckets are mounted on three wheels or DRUMS, of increasing diameter toward the exhaust end. This increase is made in order to avoid the use of very long buckets (long radially) toward the low pressure end. The valve V is employed to admit high-pressure steam to the lower pressure stages. This is done when overloads are encountered

but involves some loss of efficiency, there being an impaired relation of velocities. The three short drums to the right of the steam inlet E may be provided with reversed buckets, then constituting an ASTERN turbine. Economy is not necessary when moving astern: hence this system contains only a few stages. The reversing action is not considered in the velocity and pressure curves below the illustration. Means must be provided for directing the steam either to the right or to the left from E. Values of u for the various drums have ratios equal to those of drum diameters. A common successive ratio is $\sqrt{2}$. The work done by the successive drums increases toward the exhaust end. With three drums, the relative amounts of work are often in the ratio $1 : 1 : 1\frac{1}{2}$.

Suppose the adiabatic drop to be as in Art. 144, 317 B.t.u. Bucket friction should be low because of the low steam speeds used, but high because of the large number of buckets. There is a leakage loss past the blade tips, which reduces efficiency. Allowing for leakage and rotation loss (Art. 141), the work at the buckets may be expected to be around 75 per cent. of the adiabatic heat drop in carefully designed machines.

Assume the probable work at the buckets to be in our case 230 B.t.u., and that 60 B.t.u. are to be derived from a first drum. How many stages (rows of buckets) shall that drum have?

Assume $u = 100$, $\alpha = 26°$, initial steam velocity 257 ft. per sec. By the method of Fig. 57, the work done in the first stage is 1.45 B.t.u. It will be greater in later stages. The number of stages will therefore be less than $60 \div 1.45$, or less than 41.

Assume a probable variation of steam velocities from beginning to end of drum as in Fig. 56. In general, this velocity increases more and more rapidly toward the low-pressure end, as here indicated. The curve may be represented by an equation in the form

$$U = a \times e^{bs}, \tag{83}$$

where U = steam velocity, ft. per sec.,

s = the number of the stage,

e = base of Napierian system of logarithms, and a and b are constants.

The curve is established when the steam velocities at two stages are known. We have assumed $U = 257$ when $s = 1$. Assume also that $U = 364$ when $s = 30$. Then

$$\frac{U_{30}}{U_1} = \frac{e^{30b}}{e^b} = e^{29b}, \quad \log_e \frac{364}{257} = 29b, \quad b = 0.012.$$

$U_1 = 257 = ae^{0.012}$, $\log_e 257 = \log_e a + 0.012$, $a = 254$. Then the equation of the whole curve is $U = 254 \times e^{0.012s}$.

Referring to Fig. 57, the work done at any stage is (Equation (82)),

$$W = \frac{\bar{c}f^2}{25\,028} = \frac{\bar{f}g^2 - \bar{c}g^2}{25\,028} = \frac{b\bar{g}^2 - \bar{c}g^2}{25\,028}$$
$$= \frac{(U\cos\alpha)^2 - (U\cos\alpha - u)^2}{25\,028} = \frac{2uU\cos\alpha - u^2}{25\,028}.$$

If U = velocity at any stage, U_1 = velocity at first stage, $c = U_1/u$, then

$$\frac{U}{U_1} = \frac{e^{bs}}{e^b} = e^{b(s-1)} \quad \text{and} \quad U = cu \times e^{b(s-1)}.$$

Then

$$W = \frac{2cu^2\cos\alpha \times e^{b(s-1)} - u^2}{25\,028} = \frac{u^2}{25\,028}(2c\cos\alpha \times e^{b(s-1)} - 1)$$

The work of the whole drum is

$$\Sigma W = \frac{u^2}{25\,028}\left(2c\cos\alpha \int_1^s e^{b(s-1)}\,ds - \int_1^s ds\right)$$
$$= \frac{u^2}{25\,028}\left\{2c\cos\alpha\left(\frac{e^{b(s-1)} - 1}{b}\right) - s\right\}. \quad (84)$$

For the conditions assumed,

$$\Sigma W = \frac{10\,000}{25\,028}\left\{2 \times 2.57 \times 0.8988\left(\frac{e^{0.012(s-1)} - 1}{0.012}\right) - s\right\}.$$

To find the value of $e^{0.012(s-1)}$, put this quantity equal to x. Then $\log_e x = 0.012(s - 1)$,

$$\Sigma W = 0.4\left(4.62\frac{x-1}{0.012} - s\right) = 60,$$
$$\frac{4.62(x-1)}{0.012} = 150 + s = 385(x - 1),$$
$$\log(535 + s) - 0.00522s = 2.581.$$

By trial and error, s is found to be between 33 and 34. A whole number must be chosen. Select 34. Then in Equation (84),

$$\Sigma W = 0.4\left(4.62\frac{e^{0.396}-1}{0.012}-34\right)$$
$$= 0.4(186.3-34) = 60.92 \text{ B.t.u.},$$

instead of 60, as originally desired. The steam velocity at the 34th stage may now be computed. It will be the initial steam velocity of the next succeeding drum.

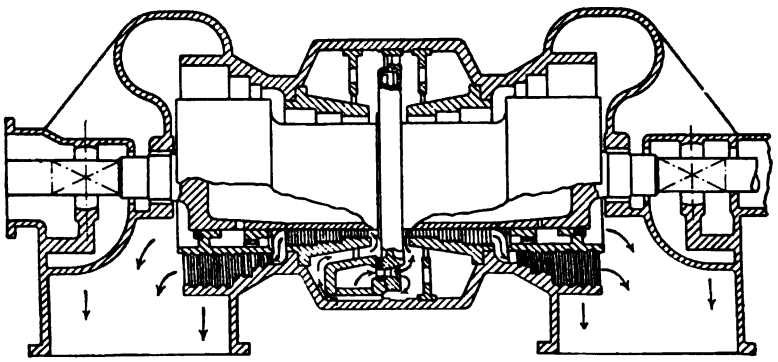

FIG. 59. Westinghouse-Parsons Double Flow Turbine.

The whole rotor of a reaction turbine is subject to an end thrust due to the static pressure of the steam toward the low-pressure end. This is often counterbalanced by "dummy pistons" on which the areas are equivalent to the exposed drum areas while steam pressures are in the opposite direction. End thrust can also be eliminated, as in Fig. 59, by supplying steam near the middle of the length and allowing it to flow both ways. This illustration shows another common feature, the use of an impulse wheel (one set of nozzles, two velocity stages) preceding the reaction elements. A large proportion of the heat is converted into work at the impulse wheel, at a point where the steam is fairly dry and the rotation loss (Art. 141) consequently small. Hence the reaction part of the turbine is reduced in size and contains fewer stages. Furthermore, tip

leakage (which is most objectionable at the high-pressure end of the reaction turbine) is limited to the low-pressure reaction stages.

Prob. 288. Check the following values of Art. 146: 1.45, 0.012, 254. What would be the work of the drum if 33 stages were used? (Ans., 58.9 B.t.u.) What is the steam velocity at the 34th stage? (Ans., 382 ft. per sec.)

Turbine Economy

147. **Low Pressure Turbines.** The turbine works with complete expansion while the engine does not. The former can do a great deal with steam rejected by an engine. Thus, in Fig. 26, suppose steam at 5 lb. pressure, 0.80 dry, to be available. Its heat content is 933 B.t.u. Expanded adiabatically to 1 lb. pressure, its content becomes 849 B.t.u. The ideal work is then 84 B.t.u. per lb. of steam, and the actual work might easily be 60 per cent. of this.

Prob. 289. A steam engine receives dry steam at 270 lb. pressure and expands it to 5 lb. pressure. A turbine then expands it down to $\frac{1}{2}$ lb. pressure. Ideally, what proportion of the total power is derived from the turbine? What is the ideal steam rate for the combination? (Ans., the turbine gives 42 per cent. of the power of the engine. Steam rate, 6.6 lb. per Ihp.-hr.) What would have been the ideal rate of a turbine alone, using dry steam at 270 lb. pressure and expanding to $\frac{1}{2}$ lb. back pressure? (Ans., 6.6 lb.)

148. **Tests of Steam Turbines.** Following are ranges of steam rates from many trials:

		Lbs. Steam per Brake hp.-hr.	Vacuum, in.
Saturated steam:	Simple impulse type	14 to 23	25 to 28
	Pressure compounded	16 to 20	27 to 29
	Reaction type	14 to 15	27 to 28
Superheated steam:	Simple impulse type	12½ to 15½	27 to 28
	Pressure compounded	12½ to 16½	28 to 29½
	Reaction type	11½ to 14½	27 to 29

Many recent tests show improvement on these figures, which in no case involved very high superheat.

Normal atmospheric pressure is 14.696 lb. per sq. in. or 29.92 in. of mercury. Vacuum is measured downward from atmospheric pressure. If the vacuum gauge reading is p_0 in. of mercury the absolute pressure is $(14.696/29.92)(29.92 - p_0) = 0.491\ (29.92 - p_0)$ lb. per sq. in.

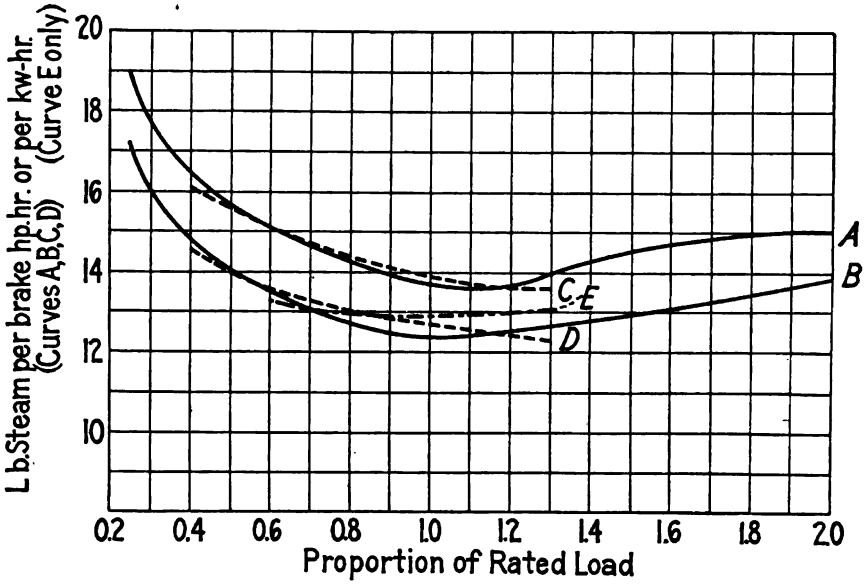

FIG. 60. Turbine Efficiency as Affected by Load.

Fig. 60 shows the effect of variable load on turbine economy, the curves being similar to those for steam engines (Fig. 29). Particulars follow:

A and *B*, 1050 hp. Westinghouse-Parsons machine, 165 lb. pressure, 28 in. vacuum. *A*, saturated steam; *B*, steam superheated 100°.

C and *D*, 400 hp. Westinghouse turbine, 28 in. vacuum: *C*, saturated steam; *D*, steam superheated 90°.

E, 9000 kw. 5-stage Curtis turbine, 215 lb. pressure, 125° superheat, 29 in. vacuum.

Fig. 61 illustrates the effect of varying superheat. The steam rate tends to bear a straight line relation with the amount of superheat. The lines are,

A, 2000 hp. Curtis turbine.

B and *C*, average results: *B* for 26 in. vacuum, *C* for 28 in. vacuum.

D and *E*, 400 hp. Westinghouse machine, 28 in. vacuum. *D*, 1.3 full load; *E*, full load.

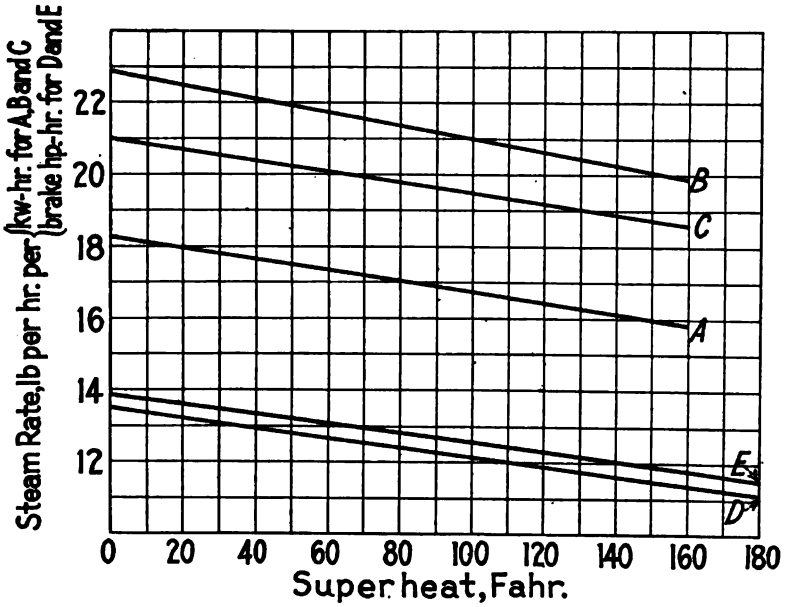

FIG. 61. Turbine Efficiency as Affected by Superheat.

Fig. 62 shows a straight line relation between vacuum and economy. *A* is a 2000 hp. Curtis machine; *B*, a 4750 kw. turbine; *C* and *D* are 1500 kw.: *D* is at $\frac{1}{3}$ load, *C* at full load.

Prob. 290. A recent turbine is guaranteed to use not over 13 lb. of steam per kw.-hr. with steam at 300 lb. pressure, superheated 300° and at 28.9 in. vacuum. Are there any rates shown in Art. 148 or Figs. 60–62 indicating that this is possible? What is the ideal steam rate per hp.-hr. under these steam conditions? (Ans., 5.44 lb.) Per kw.-hr.? (Ans., 7.3 lb.) If the efficiency from shaft to switchboard is 0.93, the guarantee implies what steam rate per brake hp.-hr.? (Ans., 9 lb.) If the mechanical efficiency is 0.95, what is the relative efficiency? (Ans., 0.64.)

Prob. 291. Find the vacuum in in. of mercury at normal atmospheric pressure when the back pressure is 1 lb. absolute. (Ans., 27.88 in.)

Prob. 292. The barometer reads 29.4 in. The vacuum gauge reads 27 in. What is the absolute back pressure? (Ans., 1.18 lb.)

Prob. 293. In Fig. 60, what percentage reduction in steam rate results at normal load by substituting for saturated steam, steam superheated 90° or 100°? (Ans., 8.6 and 9.5 per cent.)

Prob. 294. What is the weight of steam used per hour at 100 per cent. overload by turbine *B*, Fig. 60? (Ans., 29,000 lb.)

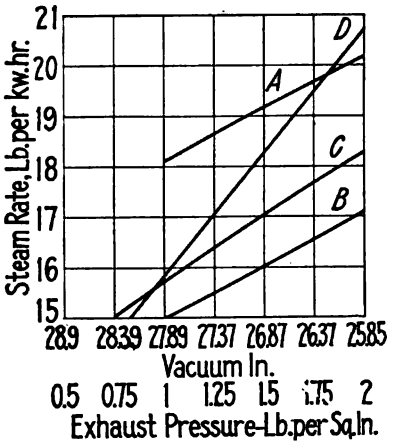

FIG. 62. Turbine Efficiency as Affected by Vacuum.

Prob. 295. In Fig. 60, which is the most economical turbine at normal load? At loads varying from 60 to 140 per cent. of rated capacity?

Prob. 296. In Fig. 61, state for all cases the percentage saving in weight of steam due to superheating 100°. (Range is from 7 to 11 per cent.)

Prob. 297. In Fig. 61, with dry steam, what is the percentage reduction in steam rate by increasing the vacuum from 26 to 28 in.? What is it when the steam is superheated 100°? 160°? (Ans., 8, 7, 6½ per cent.)

Prob. 298. State in words the significance of the crossing of curves *C* and *D*, Fig. 62.

CHAPTER IX

INTERNAL COMBUSTION ENGINES

Four-Cycle Engines with Explosive Charge

149. **The Otto Cycle.** Fig. 63 is an indicator diagram from a gas engine. Note that it represents FOUR successive strokes. At the point *c*, the piston is at the left-hand end of its stroke, the exhaust valve is closed and the inlet valve is open. The piston moves one full stroke to the right, drawing in a charge of

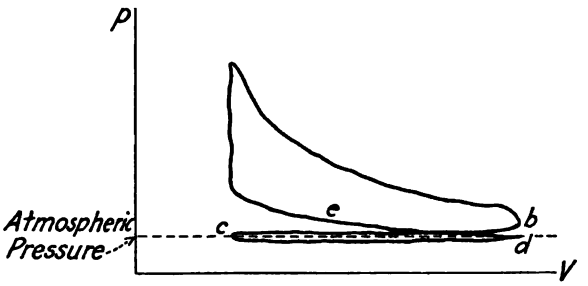

FIG. 63. Gas Engine Indicator Diagram.

COMBUSTIBLE MIXTURE. This operation is represented by the line *cd*, along which the pressure is slightly below that of the atmosphere.

At *d*, the inlet valve closes. The piston now makes a stroke to the left, COMPRESSING the combustible mixture along *de*· · ·. At or near the end of this stroke, the charge is ignited by appropriate means, and there is a rapid combustion accompanied by a sudden rise of pressure. The piston then moves to the right under the expansive action (falling pressure, increasing volume) of the heated gas. At or near the end of this stroke, the exhaust valve opens and the burnt mixture is discharged (line *bc*) at a pressure slightly above that of the atmosphere. Fig. 65 shows the engine.

150. **Ideal Cycle.** This operation is idealized in Fig. 64, diagram *abcd*. Here the counter-clockwise (power-CONSUMING) area due to intake and exhaust is disregarded, the combustion is assumed to be perfectly instantaneous and compression and expansion are taken to be adiabatic. The cycle is therefore bounded by two adiabatics and two lines of constant volume.

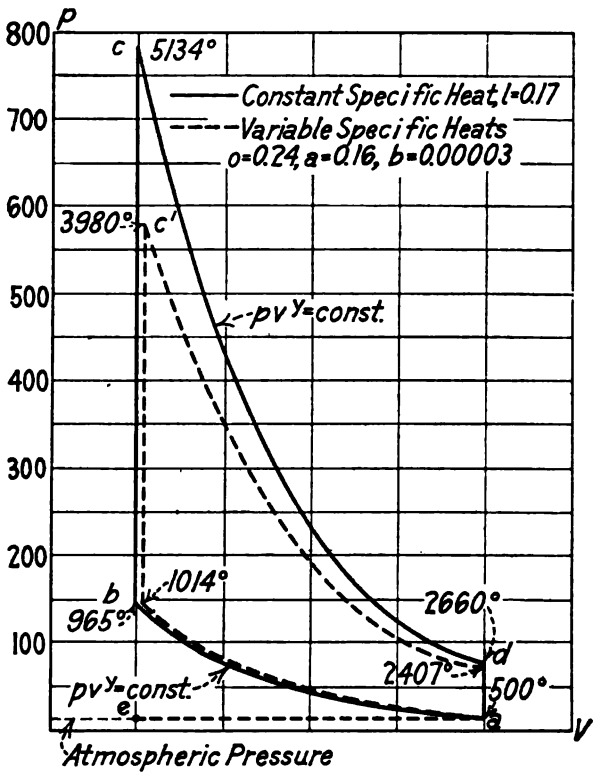

FIG. 64. Ideal Otto Cycle.

Remembering that efficiency is (HEAT ABSORBED — HEAT REJECTED) ÷ HEAT ABSORBED, and that there is neither absorption nor rejection of heat along an adiabatic,

$$e = \frac{H_{bc} - H_{da}}{H_{bc}} = 1 - \frac{H_{da}}{H_{bc}} = 1 - \frac{wl(T_d - T_a)}{wl(T_c - T_b)} = 1 - \frac{T_d - T_a}{T_c - T_b},$$

the specific heat at constant volume being denoted by l and the weight of mixture by w. Now the paths cd and ba are isodiabatics (Art. 31). Hence $T_d/T_a = T_c/T_b$,

$$\frac{T_d}{T_a} - 1 = \frac{T_c}{T_b} - 1, \quad \frac{T_d - T_a}{T_c - T_b} = \frac{T_a}{T_b},$$

$$e = 1 - \frac{T_a}{T_b} = \frac{T_b - T_a}{T_b} = 1 - \left(\frac{1}{C}\right)^{(y-1)/y} = 1 - \left(\frac{c+1}{c}\right)^{1-y}, \quad (85)$$

if the mixture is regarded as a perfect gas. In these expressions, C = COMPRESSION RATIO = p_b/p_a; c = CLEARANCE = $V_b/(V_a - V_b)$. One of the expressions looks like that previously given (Art. 34) for the efficiency of the CARNOT cycle. It has a different meaning: T_b and T_a are NOT the extreme temperatures of the cycle. The efficiency of the Otto cycle is less than that of the Carnot cycle. It is much greater than that of the Rankine cycle, used for the steam engine and turbine: mainly because of the very great temperature range employed in gas engines. Clearance in gas engines is made to suit the compression desired. It is not to be viewed as a "necessary evil," as it is in steam engines.

Prob. 299. Given $C = 10$, find efficiency and clearance, using $y = 1.4$. (Ans., $e = 0.48$, $c = 0.24$.)

151. **Mean Effective Pressure.** Still considering the substance to be a perfect gas,

$$p_m = \frac{\Sigma W}{144(V_a - V_b)} = \frac{p_c V_c - p_d V_d - p_b V_b + p_a V_a}{(y-1)(V_a - V_b)} \text{ (see Art. 28)}$$

$$= \frac{V_c(p_c - p_b) + V_a(p_a - p_d)}{(y-1)(V_a - V_b)}.$$

But $V_c/(V_a - V_b) = c$: while

$$(V_a - V_b)/V_a = 1 - \frac{V_b}{V_a} = 1 - \left(\frac{1}{C}\right)^{1/y} = 1 - C^{-(1/y)}.$$

Then, remembering that $p_a = p_b/C$ and $p_d = p_c/C$,

$$p_m = \frac{c(p_c - p_b)}{y - 1} + \frac{p_b - p_c}{C(1 - C^{-(1/v)})}$$

$$= \frac{p_c - p_b}{y - 1} \cdot \left(c - \frac{1}{C - C^{(v-1)/v}} \right). \quad (86)$$

A more useful expression is

$$p_m = \frac{778\Sigma H}{144(V_a - V_b)} = \frac{5.403wl(T_c - T_b)e}{V_a - V_b}.$$

This has a special meaning. The quantity $wl(T_c - T_b)$ represents the heat available in the cylinder for raising the temperature of w lb. of mixture, at constant volume, from T_b to T_c.

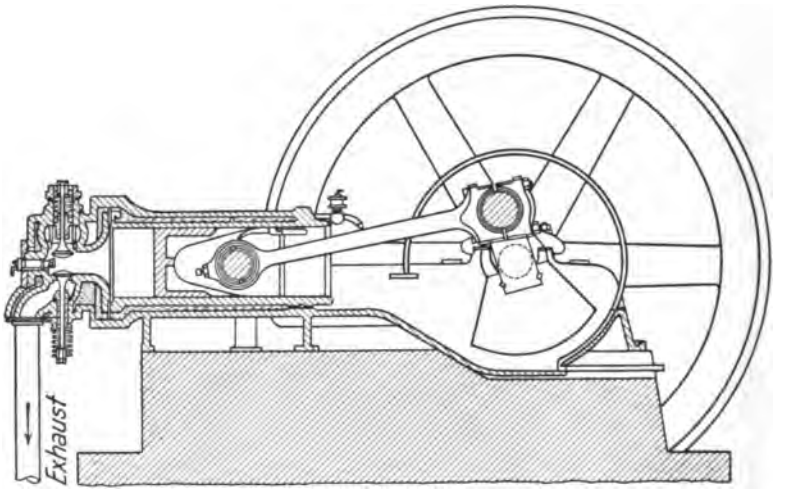

FIG. 65. Otto Cycle Gas Engine.

It is the HEAT VALUE of the mixture present. All of that heat, under ideal conditions, is used to increase the temperature along *bc*. Now consider the hitherto ignored intake stroke, *ea*, Fig. 64. At the beginning of this stroke, the volume of exhaust gas in the cylinder was V_c. At the end of the intake stroke, the volume of exhaust gas and fresh mixture was V_a. Hence the volume of fresh mixture admitted, and present during combustion, was $V_a - V_e = V_a - V_b$. Then THE HEAT VALUE

OF 1 CU. FT. OF MIXTURE SUPPLIED was $wl(T_c - T_b) \div (V_a - V_b)$. Call this B. Then

$$p_m = 5.403Be, \text{ lb. per sq. in.} \tag{87}$$

Here e = efficiency, from Equation (85).

Prob. 300. In Prob. 299, find p_m from Equation (86) if $p_a = 14.696$, $p_c = 346.96$. (Ans., 58 lb.) What heat value per cu. ft. of mixture does this imply? (Ans., 22 B.t.u.)

152. **Variable Specific Heat.** At the high temperatures prevailing in the cylinder of an internal combustion engine, the specific heats are highly variable, although the gases are not so imperfect as to depart from Equation (12). Some refinement is therefore necessary over the foregoing.

Suppose $l = a + bT$, where a and b are constants. Then

$$H_{bc} = \int_b^c l dT = \int_b^c a dT + \int_b^c bT dT$$

$$= a(T_c - T_b) + \frac{b}{2}(T_c^2 - T_b^2).$$

Similarly, $H_{ad} = a(T_d - T_a) + (b/2)(T_d^2 - T_a^2)$, and

$$e = \frac{H_{bc} - H_{ad}}{H_{bc}} = 1 - \frac{H_{ad}}{H_{bc}}$$

$$= 1 - \frac{a(T_d - T_a) + \frac{b}{2}(T_d^2 - T_a^2)}{a(T_c - T_b) + \frac{b}{2}(T_c^2 - T_b^2)}. \tag{88}$$

Under these conditions, adiabatics are not isodiabatics, for the equation of the adiabatic is not $pV^v = \text{const.}$ On the contrary, if we may write $k = o + bT$, $R/778 = R' = k - l = o - a$, the equation of the adiabatic (with variable specific heats) is thus derived:

In an adiabatic process, the external work done equals the loss of internal energy: or $T = -W$ (Art. 28). Applying this to a very brief process, during which the pressure is substantially constant, $ldT = -Pdv/778$ for 1 lb. weight. Writing $l = a + bT$, $P = RT/v$, this may be reduced to

$$\frac{adT}{T} + bdT = -\frac{R'dv}{v}.$$

Now differentiate the perfect gas equation, $Pv = RT$. This gives $Pdv + vdP = RdT$. Dividing by Pv,

$$\frac{dv}{v} + \frac{dP}{P} = \frac{dT}{T}.$$

Substituting this value in the first term of the preceding equation, we obtain

$$(a + R')\frac{dv}{v} + a\frac{dP}{P} + bdT = 0.$$

Integrating, $(a + R') \log_e v + a \log_e P + bT = \text{const.}$

$$\frac{o}{a}\log_e v + \log_e P + \frac{b}{a}T = \text{const.},$$

$$Pv^{o/a}e^{bT/a} = \text{const.}, \tag{89}$$

where e is the base of the Napierian system of logarithms.

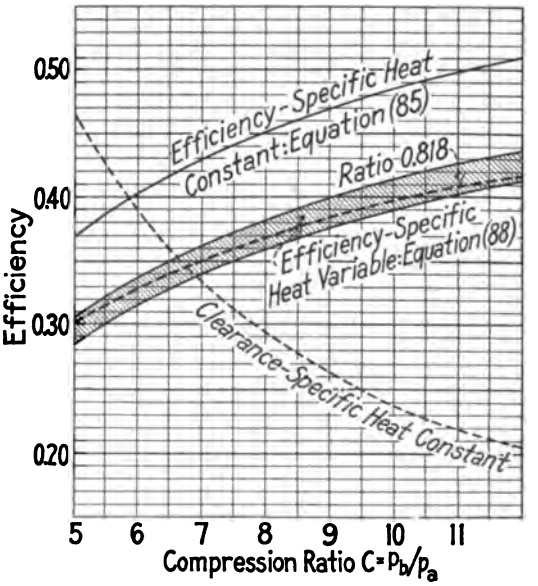

FIG. 66. Otto Cycle Efficiencies, Constant and Variable Specific Heats.

In the ideal case, the mixture is considered to be pure air. It is, in fact, mostly air. Values of a, b and o are not exactly known: but approximately, $a = 0.1801$, $b = 0.0000283$, $o = 0.2511$. Fig. 64 shows by dotted lines the resulting cycle, to scale, though not for these particular values of the constants. In both diagrams, $p_a = 14.696$, $T_a = 500$, $C = 10$, and the heat evolved during combustion is 700 B.t.u. per lb. of mixture. The chief difference is with respect to the rise of temperature along bc. With $l = 0.17$, this is $700 \div 0.17 = 4120°$.

Fig. 66 compares efficiencies of the two cycles, the broad band of results from Equation (88) covering the whole range of proposed values of a, b and o. The heavy dotted line within this band represents an efficiency which is 81.8 per cent. of that by Equation (85). This line may be regarded as representing results of Equation (88) as closely as they can be known at present. In other words, the ideal efficiency is 81.8 per cent. of that given by the simple (constant specific heat) expression of Equation (85).

Prob. 301. In Art. 152, show that the heat absorbed along bc' is approximately 700 B.t.u. Take values of constants from Fig. 64.

Prob. 302. From Equation (89), show that

$$\log_e \frac{T_b}{T_a} + \frac{b}{a}(T_b - T_a) = \frac{a - o}{a} \log_e \frac{V_b}{V_a},$$

$$\frac{o}{a} \log_e \frac{T_b}{T_a} + \frac{b}{a}(T_b - T_a) = \frac{o - a}{a} \log_e \frac{p_b}{p_a}.$$

Prob. 303. Does Charles' law apply when the specific heats are variable as in Art. 152?

Prob. 304. What is the probable true ideal efficiency when $C = 10$? (Ans., 0.397.)

153. **Variations of Ideal Efficiency with Load: Governing.** When the load on an engine decreases, the supply of heat should be decreased. This is accomplished in a constant speed engine one of four ways in:

(1) Vary the time of ignition. This simply throws away heat not needed.

(2) Omit drawing in a charge for one or more strokes ("hit or miss"). This leads to irregular speed and is applicable only to small engines.

(3) Dilute the charge with an excess of air.

(4) Decrease the weight of charge drawn in.

Methods (3) and (4) are important. We have seen that efficiency depends on compression (Art. 150, Equations (85), Fig. 66): strictly, on the COMPRESSION RATIO, C. Compression is limited by the temperature attained at b, Fig. 64. We must

TYPICAL IGNITION TEMPERATURES, °F.*

Ether, 750	Acetylene, 760–820	Benzene, 780	Alcohol, 950
Liquid Fuels, 950–1020	Equal Parts H and O, 957		Benzol, 970
Ethylene, 1005–1015		Hydrogen, 1075–1100	
Illuminating Gas, 1100		Methane, 1200–1240	

not reach the ignition temperature by compression, else pre-ignition (self-ignition) will occur. For each possible mixture, there is a limiting permissible compression and hence a limiting efficiency. We should work as close to this compression as we can. In fact, without compression it would be difficult or impossible to ignite some fuels at all. The accompanying table lists the more common mixtures in the approximate order of allowable compressions, and therefore in the approximate order of efficiencies. (The values of the last column are discussed later.)

Now if we dilute the charge with excess of air, we raise its ignition temperature (and also make ignition occur more slowly). Hence we are operating at an efficiency below the best possible efficiency.

If we decrease the weight of charge, we also dilute it. This would occur, if for no other reason, because the charge is mixed with an approximately constant quantity of exhaust gas left

* Except as noted, in air at normal pressure.

in the clearance space. Here also we are operating, when at reduced load, at less than maximum possible efficiency.

Usual Compression and Mean Effective Pressures for Various Fuels*

Fuel and Engine	Compression Pressures		p_m'
	Usual Range	Average	
Kerosene in small hot bulb engines	45– 90	75	50– 80
Gasoline, automobile	60–110	80	70– 90
Kerosene, carburettor engine	60–100	80	50– 75
Fuel oil, hot bulb engine	—	60	50– 85
Gasoline, stationary and marine engines	75–120	85	80– 95
Heavy fuel oils	75–135	—	50– 70
Rich gas mixtures	70– 90	—	70– 85
Lean gas mixtures	90–135	—	65– 80
Natural gas, large engines	90–175	135	80–110
Coal gas, small engines	90–135	115	60–100
Carburetted water gas, small engines	90–120	105	60–100
Producer gas, general	115–175	145	55– 90
Coke oven gas, large engines	120–150	135	80–110
Blast furnace gas, very large engines	135–205	170	80–120
Alcohol, carburettor engine	135–225	165	130

In either case, the efficiency at light loads is less than it should be, and (actually) much less than at full load. Methods (3) and (4) for governing are generally (and unavoidably) used simultaneously. When the mixture is diluted by throttling the gas, the intake suction is increased and the pressure at *a*, Fig. 64, is reduced. The weight is therefore reduced. When the weight of charge is decreased, the increased intake suction modifies the proportions of the charge.

An important defect in gas engine governing arises from the fact that the governing action (by methods 2, 3 or 4) is exerted one full revolution before its effect is felt.

Governing by dilution of charge naturally decreases the B.t.u. per cu. ft. of mixture and hence the value of p_m (Art. 151). Governing by decreasing weight of charge reduces the B.t.u. PER CU. FT. OF DISPLACEMENT and hence the value of p_m.

Prob. 305. What is the best ideal efficiency to be expected with producer gas, if the pressure at *a*, Fig. 64, is 14.6 lb.? (Ans., 0.417.)

* Mostly from Lucke.

Prob. 306. State an average value of C for alcohol, if $p_a = 14.35$. (Ans., $11\frac{1}{2}$.)

Prob. 307. In an engine governed by varying the weight of charge, how would the efficiency vary with the load if the CLEARANCE were perfectly adjustable?

154. **The Actual Engine.** In the actual engine, the intake pressure is somewhat below atmospheric and $p_a = 8\frac{1}{2}$ to $14\frac{1}{2}$. The compression is not precisely adiabatic. It may be represented by $pv^n = \text{const.}$, with n between 1.25 and 1.35. Ignition is not instantaneous and the combustion line has a perceptible slope to the right as it rises. This is accentuated at high piston speeds or with slow ignition (improper mixture) conditions. The speed of flame propagation is of the order of 7 to 20 ft. per sec. at the pressures existing and with correct mixtures. The maximum temperature and pressure are much less than those computed as for Fig. 64 by either method. This is partly because the volume increases during combustion and partly because of the cooling which occurs. The maximum temperature is rarely much over 3000° F. The expansion curve is a polytropic with n between 1.3 and 1.5. The temperature at the point when the exhaust valve opens may be anywhere from 1200° to 2000°. The exhaust valve opens considerably before the end of the stroke. The average exhaust pressure is 16 or 17 lb. The temperature in the exhaust pipe is 700° to 1000°.

155. **Diagram Factor and Volumetric Efficiency.** The actual indicated thermal efficiency is usually from 40 (occasionally down to 30) to 55 per cent. of that given by Equation (85), or 49 to 67 per cent. of that by Equation (88), for the compression used. The "relative efficiency" (Art. 93) is therefore about the same as for steam engines: but its variations have not been as thoroughly studied. The lower values of relative efficiency are obtained with engines employing carburettors to vaporize the fuel. This percentage which the actual indicated thermal efficiency bears to the ideal efficiency is also called the DIAGRAM FACTOR.

The VOLUMETRIC EFFICIENCY is the ratio of the volume of gas drawn in (measured at normal barometer and 62°) to the corresponding piston displacement. In discussing the ideal cycle, Art. 151, it was considered to be 1.0. In practice, it is decreased by the excess of the exhaust pressure over 14.696, by the suction below that amount, and by any intake temperature above 62°. Values of the volumetric efficiency are $e_v = 0.50$ to 0.87. It is lowest in high speed engines and with automatic valves. It is particularly low in airplane engines when flying at high altitudes. SUPERCOMPRESSION involves filling the cylinder by means of a special external pump: by which method e_v may be greatly increased.

The actual mean effective pressure is then to be obtained from a modification of Equation (87):

$$p_m' = 5.403 f B e e_v, \tag{90}$$

where B = B.t.u. per cu. ft. of mixture, f = diagram factor, e = ideal efficiency. Values of p_m' are given in the second table of Art. 153. The ratio of actual to ideal mean effective pressure (p_m'/p_m) is about equal to the ratio of actual and ideal pressure rises, $p_c - p_b$, Fig. 64.

Prob. 308. What maximum mean effective pressure may be expected with blast furnace gas (60 B.t.u. per cu. ft. of mixture) in a large slow-running engine with mechanically operated valves? (Ans., 87 lb.)

Prob. 309. An automobile engine gives 65 lb. mean effective pressure. It has average compression. Assume 100 B.t.u. per cu. ft. of mixture. Make an estimate of its volumetric efficiency. (Ans., 0.82.)

Prob. 310. Compute B for blast furnace gas, from the following table. (Ans., 57.)

Prob. 311. Assume gasoline to be represented by C_6H_{14}, and to contain 19 600 B.t.u. per lb. What is its heat value per cu. ft. OF VAPOR at standard conditions (32°, 14.696 lb.)? Compute B. (Ans., 4710 and 96 B.t.u.)

Properties of Gas Fuels (Lucke)

	Composition by Volume							B.t.u. per Cu. Ft.*
	CH_4	C_6H_6	CO	O_2	H_2	CO_2	N_2	
Natural gas (Kansas)	0.982	0.001	0.0025	0.0025	—	—	—	950.1
Coke oven gas	0.343	0.040	0.0600	0.0110	0.420	0.025	0.101	579.8
Retort coal gas	0.3135	0.022	0.0860	0.0035	0.525	0.015	0.035	542.6
Oil gas	0.525	0.235	0.0100	0.0050	0.185	0.005	—	1196.5
Blast furnace gas	0.002	—	0.2861	—	0.027	0.114	0.571	107.5
Water gas	0.044	0.001	0.4485	0.0050	0.456	0.045	0.001	329.9
Producer gas (from hard coal)	0.002	—	0.2610	0.0020	0.150	0.053	0.532	134.7

156. **Mechanical Efficiency: Overload Capacity.** If we follow steam engine practice, the mechanical efficiency is e_M = bhp. ÷ ihp., where ihp. refers to the NET diagram of Fig. 63; i.e., to

* Low value, i.e., heat evolved when the products of combustion are NOT condensed in the calorimeter: taken at 32° and normal barometer. NOTE: the values given in the last column are NOT those of B, Equation (90). That symbol stands, not for the heat value of the fuel, but for the heat value of 1 cu. ft. of combustible mixture, formed by adding the proper amount of air to the fuel. The weight of chemically ideal mixture is computed by utilizing the law that COMBINING PROPORTIONS BY VOLUME ARE THOSE OF THE NUMBER OF MOLECULES ENTERING INTO THE REACTION. Thus CO and H_2 require half their volume of oxygen, C_2H_4 three times its volume of oxygen, CH_4 twice its volume and C_6H_6 seven and one half times. From the total oxygen requirement thus computed deduct the oxygen in the fuel and multiply the remainder by 4.75 to obtain the volume of AIR which must be added to 1 cu. ft. of fuel to produce a perfect mixture. Thus, for natural gas:

$$
\begin{aligned}
&0.982 \times 2 = 1.964\\
(C_6H_3)\ &0.001 \times 7\tfrac{1}{2} = 0.0075\\
&0.0025 \times \tfrac{1}{2} = 0.0013\\
&\qquad\qquad 1.9728\\
&\qquad\qquad 0.0025\\
&1.9703 \times 4.75 = 9.37\\
&\qquad\qquad\quad 1.0\\
&950.1 \div 10.37 = 92 \text{ B.t.u. per cu. ft. mixture at } 32^\circ \text{ F.}
\end{aligned}
$$

At 62° and normal barometer,

$$B = 92 \times \frac{460 + 32}{460 + 62} = 86\tfrac{1}{2} \text{ B.t.u.}$$

The mixture obtained in practice is never exactly right. Hence B is less than this. If the mixture is far from correct, combustion is slow. If very weak or very strong, the mixture may refuse to ignite at all.

the clockwise cycle MINUS the negative loop. It is logical, however, to regard the ihp. as the larger value due to considering the clockwise cycle alone, without deduction for the loop, because that loop is due to friction. The friction is of course fluid friction rather than mechanical friction. The loop consumes 5 to 10 per cent. of the power of the engine. If we regard the clockwise diagram as representing the ihp., the mechanical efficiency ranges from 0.75 to 0.85, increasing for large engines and being higher for double-acting than for single-acting engines (Art. 7). Let A = gross ihp., C = loop ihp., D = brake hp.: then as now defined $e_M = D/A$ whereas as defined for a steam engine

$$e_M = D/(A - C).$$

Equation (90) shows that p_m' varies with f, B, e and e_v. Naturally, f and e_v will be kept as high as possible at the rated load condition. The value of e depends on the compression alone, and that will be as high as the nature of the fuel will permit. The effort will be made to have the mixture proportions right. Hence all factors have maximum values at rated load. In a constant speed engine, there is consequently no way of increasing power beyond that for which the engine was built. Efficiency and output vary in the same direction, not at all as in a steam engine. Maximum efficiency occurs at maximum power. An Otto cycle engine has, strictly speaking, no overload capacity. A margin of power for emergencies can be provided only by rating the engine at a power less than that obtainable at maximum efficiency (Art. 165).

Prob. 312. The gross ihp. is 100, loop hp. = 8, brake hp. = 80. Find two values of e_M. (Ans., 0.87 and 0.80.) Which is adopted in the text? (Ans., 0.80.)

Prob. 313. Sketch a curve of thermal efficiency against brake hp. Compare Fig. 71.

Other Otto Cycle Engines

157. **Two-Cycle Engine.** The action of the two-cycle engine is suggested (and somewhat exaggerated) in Fig. 67. The

exhaust opens at *d*, quite early in the expansion stroke. When the pressure has fallen to p_e, the inlet opens to admit a charge of mixture UNDER PRESSURE. This flows in along *efg*. The exhaust does not close until point *a* is reached. Compression then begins. A separate pump or its equivalent is provided for

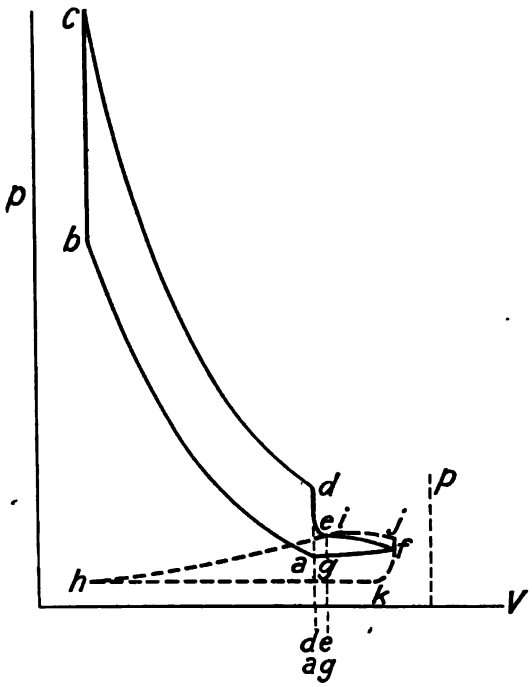

FIG. 67. Two-Cycle Engine.

delivering the mixture to the cylinder. Its indicator diagram, shown dotted and with the V axis reversed, corresponds with the negative loop of Fig. 63: but the loop now consumes 7 to 12 per cent. of the engine power and the mechanical efficiency as defined in Art. 156 is reduced. It is from 0.65 to 0.70, being high for large engines.

Along *efg*, inlet and exhaust valves are both open. Hence some fuel will be lost. This is mitigated by SCAVENGING, i.e.,

by having a blast of pure air precede the injection of the charge: but it is never entirely eliminated. The two-cycle gas engine is about 20 per cent. less efficient than the four-cycle. If it were not for its larger fluid friction loss, it would give twice the power, because only two strokes, instead of four, are necessary to complete a cycle. (Exhaust and intake are crowded into a small portion of the expansion and compression strokes.) Actually, the gain in power is about 70 or 80 per cent.

The necessary outside compression of mixture may be provided by (*a*) using the idle end of an otherwise single-acting cylinder: (*b*) utilizing the crank case as a compression chamber: (*c*) separate pumps. Methods (*a*) and (*b*) make the parts of the engine inaccessible and complicate lubrication and cooling. A two-cycle engine may be VALVELESS, the passages being slots in the cylinder wall, opened and closed by the piston.

In the smaller and cheaper two-cycle engines, e_v is particularly low. The ideal efficiency is that given by Equations (85) or (88). Diagram factors and mean effective pressures are about 0.8 those of four cycle engines.

Prob. 314. State the probable actual indicated thermal and brake thermal efficiency, and mean effective pressure, of a large slow-running two-cycle engine using blast-furnace gas. See Art. 155, Prob. 308, Prob. 310, Art. 156. (Ans., 0.25, 0.165, 63 lb.)

158. **Size and Power of Engine.** Following Art 7, for a four-cycle single-acting CYLINDER (which yields power at one-fourth the rate of a steam cylinder)

$$\text{ihp.} = \frac{\pi}{16} d^2 p_m' \times \frac{2LN}{33\,000} = \frac{\pi p_m' L d^2 N}{264\,000}$$

$$= \frac{p_m' AS}{132\,000} = \frac{p_m' LAN}{66\,000}. \quad (91)$$

For other types, use the following multipliers for the above:

2 for four-cycle double-acting,
2 for two-cycle single-acting,
4 for two-cycle double-acting.

Most engines have a plurality of cylinders, all of which develop equal power. Double-acting engines are difficult to lubricate and cool.

Prob. 315. In Prob. 314, find the diameter and stroke of an engine to develop 500 brake hp. at 600 ft. piston speed and 110 r.p.m., if two-cycle, two-cylinder, single-acting. (Ans., 23.6 by 32.7 in.)

Prob. 316. The commercial formula for brake hp. of an automobile engine is $nd^2/2.5$, where n = no. of cylinders, d = dia. in inches. If this is based on 1000 ft. piston speed and $e_M = 0.75$, for single-acting four-cycle engines, what mean effective pressure is assumed in the cylinders? (Ans., 90 lb.)

159. **Oil Engines.** Engines employing carburettors to vaporize a volatile fuel are not to be classed as oil engines, although kerosene, and some few heavier oils, may be used in such engines if the carburettor is heated. An oil engine, properly so called, draws in pure air on its suction stroke and the fuel is injected separately. A two-cycle oil engine may therefore avoid the loss of fuel to the exhaust (Art. 157) which is the largest factor in producing low efficiencies of two cycle engines.

The less volatile fuels are employed for oil engines. They cannot be ignited by electric spark. Hence such engines are provided with an uncooled "hot bulb," "hot tube" or plate, which communicates with the clearance space. This is kept red hot. It is heated by a torch for starting. It ignites the combustible mixture only after the temperature of the air in such mixture has been somewhat increased by compression.

Some oil engines inject the fuel during the suction stroke or the early part of the compression stroke. The whole charge is then heated by the compression before it actually ignites. The time of ignition is determined by the proportioning of the uncooled surface, the compression, etc., and is apt to be irregular at variable loads. In other engines, there is TIMED injection of the fuel, close to the end of the compression stroke. Ignition cannot then be premature. Further, the compression may

be carried very high and high efficiency secured. On the other hand, the fuel must be injected against a high resisting pressure. Heavy oils must be finely subdivided in order that ignition may occur. Hence the best method of delivering them to the cylinder is by a compressed air blast. The necessary air pressure for maximum subdivision of the oil particles is about twice the pressure in the cylinder (Art. 136). Hence with high compressions, very high air pressures and multi-stage air compression are necessary for feeding the fuel.

Oil engines of this type are easily governed by varying the quantity of fuel injected. The diagram factor decreases somewhat at light loads. Overloads lead to over-rich mixtures, overheating and a smoky exhaust. Compression pressures (p_b, Fig. 64) as high as 300 lb. are being used, but these lead to extremely high maximum pressures.

For liquid fuels in general, the specific gravity is

$$s = \frac{140}{130 + b},$$

where b = reading of Baumé hydrometer. Water weighs 8.25 lb. per gal. at 62°. The weight of a gallon of oil at this temperature is $8.25s$ lb. and its heat value is very nearly

$$18\,650 + 40(b - 10) \text{ B.t.u. per lb.}$$

Oils are sold by bulk. They expand 0.00055 of the original volume per degree rise in temperature. Fuel oil contains C, 0.80 to 0.87 and H_2, 0.10 to 0.14, the balance being O_2 and N_2 with (frequently) very small traces of sulphur.

Prob. 317. How much heat is contained in 50 gal. of fuel oil of 30° Baumé density at 62° F.? (Ans., 7 000 000 B.t.u.) What will be the bulk of this oil at 92° F.? (Ans., 50.83 gal.)

The Diesel Cycle

160. **Ideal Cycle.** The ideal Diesel cycle draws in a charge of pure air at atmospheric pressure (*ea*, Fig. 68) and compresses it adiabatically to about 500 lb. pressure. The temperature thus

attained is so high that upon injection of the fuel it immediately ignites. The delivery of fuel is so retarded that combustion occurs at CONSTANT PRESSURE, *bc*, hence there is no explosive action. The supply of fuel is cut off and combustion ceases at *c*.

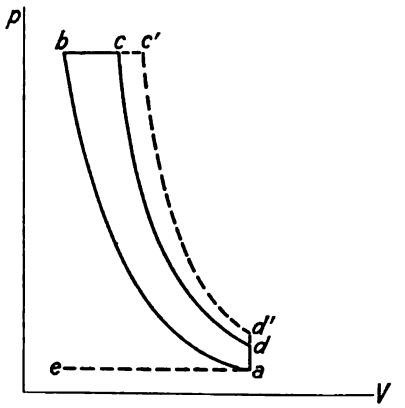

FIG. 68. Ideal Diesel Cycle.

Adiabatic expansion follows and the exhaust valve opens at *d*. The pressure drops instantaneously to that of the atmosphere, at which pressure the burnt gases are expelled.

Under these ideal conditions, the efficiency is,

$$e = \frac{H_{bc} - H_{da}}{H_{bc}} = 1 - \frac{H_{da}}{H_{bc}}$$

$$= 1 - \frac{wl(T_d - T_a)}{wk(T_c - T_b)} = 1 - \frac{T_d - T_a}{y(T_c - T_b)}. \quad (92)$$

The ideal mean effective pressure is (following Art. 151)

$$p_m = 5.403 \frac{H_{bc}e}{V_a - V_b} = 5.403Be. \quad (93)$$

where B = heat supplied per cu. ft. of piston displacement. For average fuel oil containing C 0.85, H_2 0.12, O_2 0.01, the volume of air at standard conditions required for ideal mixture is about 200 cu. ft. per lb. of fuel. Hence in the ideal cycle with $e_v = 1.0$ (Art. 155), $B = 19\,500 \div 200 = 97\frac{1}{2}$, very nearly, and $p_m = 536e$. Actual values are less than half this (Art. 162).

Prob. 318. Given $t_a = 40°$ F., $C = p_b/p_a = 34$, $T_c = 2T_b$, $y = 1.4$, find e and p_m. (Ans., 0.574, 308 lb.)

161. **Variation of Ideal Efficiency.** The Diesel cycle is more efficient than the Otto. It is rather unique among heat engine cycles, in that its EFFICIENCY INCREASES AT LIGHT LOADS. Fig. 69 plots values of e against the RATIO OF EXPANSION, $r = V_d/V_c = V_a/V_c$, Fig. 68. This is based on $t_a = 40°$ F., $C = 34$, as in Prob. 318, with $y = 1.4$. High ratios of expansion imply low mean effective pressures. Compare the diagrams *abcd* and *abc'd'*, Fig. 68. Since the ideal value of p_m varies DIRECTLY with e (Equation (93)) the underload condition must be one of (*a*) poor mixture, (*b*) low volumetric efficiency or (*c*) low relative efficiency.

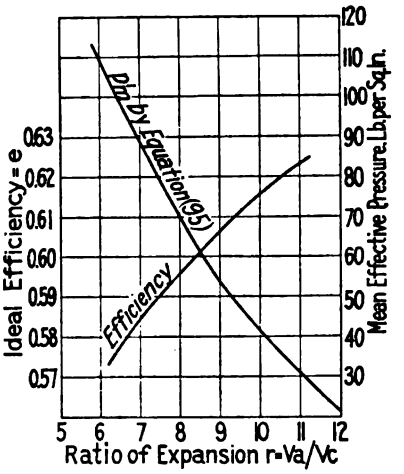

FIG. 69. Diesel Cycle at Various Loads.

162. **The Diesel Engine in Practice.** In actual engines, the efficiency occasionally, though not frequently, shows an increase with decrease of load until the load has been reduced to about half its normal value. The heat losses then begin to offset the cyclic gain, and below this point the efficiency decreases. In most cases, the efficiency falls off with decrease of load over the whole range. Most Diesel engines are four-cycle, single-acting. The two-cycle arrangement is sometimes used, requiring that the air charge be delivered at 20 to 30 lb. pressure. This consumes about 4 per cent. of the engine power. The two-cycle Diesel engine gives 70 to 80 per cent. more output than the four cycle, but its relative and actual efficiencies are about 10 per cent. less. Mechanical efficiencies of Diesel engines are from 0.70 to 0.75. The high compression used requires that the

fuel be supplied by a compressed air blast (Art. 159) at about 1000 lb. pressure. From 16 to 34 cu. ft. of free air (Art. 42) are required per brake hp.-hr., and the compressor absorbs 4 to 7 per cent. of the power of the engine.

Diagram factors in well-designed engines are around $f = 0.50$ to 0.62. The corresponding value of p_m', the actual mean effective pressure, is $5.403\, e_v f e B'$, where B' = B.t.u. per cu. ft. of combustible mixture at standard conditions. The value of p_m' may also be computed from Fig. 68,

$$p_m' = \frac{p_b(1+c)}{n-1}\left(\frac{n(1+c-cr)}{r(1+c)} - \frac{C-r^n}{Cr^n}\right), \qquad (94)$$

in which

n = exponent of polytropic curves ab and cd,
c = clearance = $V_b \div (V_a - V_b) = 1 \div (C^{1/n} - 1)$,
r = ratio of expansion = V_d/V_c,
C = compression ratio = p_b/p_a.

For the usual conditions $p_b = 500$, $C = 34$, if we take $n = y = 1.4$ and compute c as 0.088, this becomes $1360(1.4/r - 1/r^{1.4}) - 114$. The curve is plotted in Fig. 69 for $n = 1.3$ (which alters the value of c) and with which

$$p_m' = 1785\left(\frac{1.3}{r} - \frac{1}{r^{1.3}}\right) - 101. \qquad (95)$$

This last expression may be used with good results in actual engines, and there is a satisfactory reason why. It is based simply on the shape of the pV diagram, and is subject only to such error as may arise from minor irregularities or abnormally shaped curves. The cut-off at rated load is usually made to occur at about 1/10 stroke, i.e., $(V_c - V_b) = 0.10(V_a - V_b)$, Fig. 68. With the usual compression, this corresponds with a value of r around 6. Fig. 69 shows the corresponding value of p_m' to be 109. In ordinary practice the value of p_m' is from 100 to 120 lb. at rated load.

Prob. 319. Find c and r in Prob. 318. (Ans., $c = 0.088$, $r = 6.22$.)

Prob. 320. In Prob. 318, if $f = 0.50$ and $e_M = 0.75$, what weight of oil (19 500 B.t.u. per lb.) will be used per brake hp.-hr.? (Ans., 0.61 lb.)

Prob. 321. In Prob. 318, if $B' = 97\frac{1}{2}$ and $f = 0.50$, what is the volumetric efficiency when $p_m' = 100$? (Ans., $e_v = 0.66$.)

Prob. 322. If $C = 34$, $n = 1.3$, find c and r when cut-off is at 10 per cent. of stroke. (Ans., $c = 0.071$, $r = 6.26$.)

Prob. 323. In Prob. 322, find p_m' from Equation (95). (Ans., 105 lb.)

Prob. 324. With p_m' as in Prob. 323, find the dimensions of a six-cylinder four-cycle single-acting engine to develop 2400 hp. at 1000 ft. piston speed and 200 r.p.m. (Ans., diameter, 25.3 in.; stroke, 30 in.)

Tests of Internal Combustion Engines

163. **Methods of Stating Efficiency.** If Q = low heat value of fuel (Art. 155), B.t.u. per lb., W = weight of fuel used per Ihp.-hr., the ACTUAL INDICATED THERMAL EFFICIENCY is

$$e_a = \frac{2545}{WQ}.$$

If e = efficiency of ideal cycle by Equation (85), the RELATIVE EFFICIENCY is $e_R = e_a \div e$. If e_M = mechanical efficiency, the BRAKE THERMAL EFFICIENCY is $e_a e_M$ and the fuel rate per brake hp.-hr. is $W \div e_M$ lb. The INDICATED HEAT RATE is $42.42 \div e_a$. B.t.u. per Ihp.-min. The BRAKE HEAT RATE is $42.42 \div e_a e_M$, Indicator practice is less satisfactory than with steam engine, hence results are often referred to brake hp. Engines using producer gas are often reported on the basis of the FUEL RATE. If e_P = efficiency of producer and Q_P = heat value of producer fuel, B.t.u. per lb., the efficiency of the plant from fuel to cylinder is $e_P e_a$: from fuel to brake, it is $e_P e_a e_M$. The fuel rate is $2545 \div Q_P e_P e_a$ lb. per Ihp.-hr. The value of e_P is around 0.75 in good operation.

Prob. 325. In Probs. 318 and 320, state the actual indicated thermal, relative and brake thermal efficiencies, the brake fuel

rate and the indicated and brake heat rates. (Ans., 0.287: 0.50: 0.215: 0.61 lb.: 148 B.t.u.: 197 B.t.u.)

Prob. 326. An engine using producer gas, generated at an efficiency of $e_p = 0.72$, has an actual indicated thermal efficiency of 0.30 and a mechanical efficiency of 0.85. The coal used contains 12 725 B.t.u. per lb. What weight of coal is consumed per brake hp.-hr.? (Ans., 1.09 lb.)

164. **Economy Determined by Compression.** As indicated by theory (Art. 150), the efficiency at full load is chiefly a matter of compression. Good four-cycle gasoline and kerosene engines, with the low compressions prevailing for those fuels, give values of e_a from 0.15 to 0.24, the first figure being realized even in

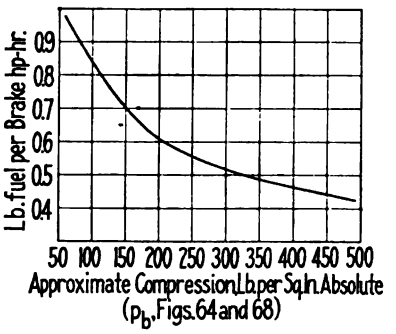

FIG. 70. Economy vs. Compression in Oil Engines.

very poor installations. With higher compressions, on producer gas, natural gas, blast furnace gas and some illuminating gases, full load values of e_a are from 0.25 to 0.32, occasionally rising to nearly 0.40. Fig. 70 illustrates guaranteed performances of a number of oil engines, including Diesel engines. A law is clearly suggested. Diesel engines attain 0.40 actual efficiency on fuel oil or kerosene. Some of the very high-pressure Otto cycle oil engines almost reach this figure.

165. **Economy in Relation to Load.** Fig. 71 shows a uniform type of curve, widely different from that obtained with steam

engines or steam turbines (Figs. 29 and 60). In all cases the efficiency is a maximum at maximum load. This applies whether governing is as in Otto cycle gas engines (Art. 153), by the Diesel variable cut-off method, or by varying speed as in the case of the Daimler engine. None of the curves for Diesel engines show the high efficiency at light load referred to in Art. 162.

Prob. 327. In Fig. 70, what is the probable brake fuel rate of an engine using 7 atmospheres compression? (Ans., 0.834 lb.) If the pressure at the beginning of compression is one atmosphere, what is the corresponding ideal efficiency by the dotted curve

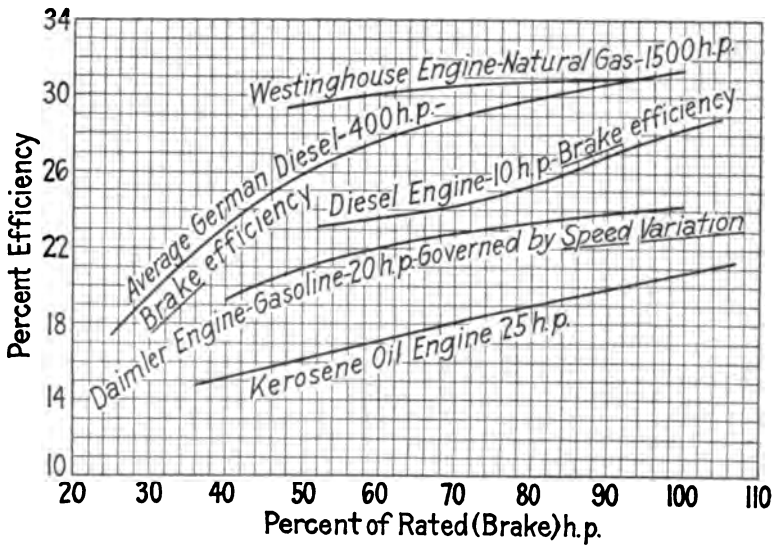

FIG. 71. Variation of Economy with Load, Internal Combustion Engines.

of Fig. 66? At a mechanical efficiency of 0.80, with fuel containing 19 500 B.t.u. per lb., what relative efficiency as compared with Equation (88) is thus implied? (Ans., 0.56.)

Prob. 328. What weight of 30° fuel oil should be used per brake hp.-hr. by the best Diesel engine of Fig. 71, at full load? At half load? (Ans., 0.42 lb.: 0.51 lb.)

Prob. 329. If the Daimler engine of Fig. 71 used 68° gasoline, how many gallons would it require per minute at 70 per cent. of its full load? (Ans., 0.0213.)

166. **Heat Balance.** The steam engine discharges the largest proportion of the heat it receives to the condenser. The gas engine divides its heat supply into three (very roughly equal) parts: one-third becomes indicated work, one-third is discharged at the exhaust, one-third is carried off by the cooling water discharged from the jackets. More accurately, indicated work accounts for 15 to 46 per cent.: jackets, for 32 to 41 per cent.: and exhaust for 24 to 33 per cent.: with a small unaccounted-for balance. The proportion to the jackets is low in very large engines.

Jacket water enters at 60° to 80° and emerges at 120° to 170°. The former exit temperature must be maintained if the fuel contains much hydrogen, in order that preignition may be avoided. Water at these discharge temperatures has very little value for heating purposes. The heat is there, but the availability is not. If e_a = efficiency of engine, j = proportion of heat supplied which is carried off by the jacket water, the heat removed by the water is $2545j \div e_a$, B.t.u. per hp.-hr. This is equal to $w_w(t_2 - t_1)$ where w_w = weight of jacket water per hp.-hr. and t_1 and t_2 are its inlet and outlet temperatures. The water pressure must be sufficient to ensure through circulation.

The exhaust heat (the amount of which may be computed in a similar way) is at high temperature (Art. 154) and is hence much more available. The transfer of this heat to water (preferably the jacket water itself) is the usually attempted method of utilizing it. This is done in a device like an economizer (Art. 124). The high gas velocity causes much difficulty in handling it, and it is a troublesome matter to avoid producing serious back pressure at the engine. Some 30 per cent. of the heat in the exhaust gases may be recovered under good conditions. If this is recovered as hot water, the best use to make of this water is for heating buildings. If the heat is recovered as steam, to be used in an engine, the very low efficiency of the steam engine enters into the matter.

Prob. 330. How much water per hour is required in the jackets of a 1000 hp. engine having an actual efficiency of 0.25 and a jacket loss of 0.35, if the water temperature rises from 70° to 140°? (Ans., 50 900 lb.)

Prob. 331. The loss to the exhaust in the above engine is 0.30, of which amount 28 per cent. is recovered. How many B.t.u. are recovered per hr.? (Ans., 853 000.) If this is all transferred to the jacket water, how much will that water be heated above 140°? (Ans., 16.8°, or to 156.8°.) If, on the contrary, it is used to generate steam at 100 lb. pressure from some of that water, how many B.t.u. must be added to each lb. of water? (Ans., 1078.4.) How many lb. of steam will be generated per hr.? (Ans., 790.) If this steam is used in a turbine which consumes 15.8 lb. of steam per Ihp.-hr., what hp. will the turbine develop? (Ans., 50.) What is the per cent. saving or gain by this complication? (Ans., 5 per cent.)

Properties of Saturated Steam

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p	t	v	h	L	H	r	n_w	n_s	n_s
0.089	32	3294	0	1073.4	1073.4	1019.3	0	2.1832	2.1832
0.363	70	871	38.06	1052.3	1090.3	994.0	0.0745	1.9868	2.0613
0.505	80	636.8	48.0	1046.7	1094.8	987.4	0.0932	1.9398	2.0330
0.696	90	469.3	58.0	1041.2	1099.2	980.8	0.1114	1.8944	2.0058
1	101.83	333.0	69.8	1034.6	1104.4	972.9	0.1327	1.8427	1.9754
2	126.15	173.5	94.0	1021.0	1115.0	956.7	0.1749	1.7431	1.9180
3	141.52	118.5	109.4	1012.3	1121.6	946.4	0.2008	1.6840	1.8848
4	153.01	90.5	120.9	1005.7	1126.5	938.6	0.2198	1.6416	1.8614
5	162.28	73.33	130.1	1000.3	1130.5	932.4	0.2348	1.6084	1.8432
6	170.06	61.89	137.9	995.8	1133.7	927.0	0.2471	1.5814	1.8285
7	176.85	53.56	144.7	991.8	1136.5	922.4	0.2579	1.5582	1.8161
8	182.86	47.27	150.8	988.2	1139.0	918.2	0.2673	1.5380	1.8053
9	188.27	42.36	156.2	985.0	1141.1	914.4	0.2756	1.5202	1.7958
10	193.22	38.38	161.1	982.0	1143.1	910.9	0.2832	1.5042	1.7874
11	197.75	35.10	165.7	979.2	1144.9	907.8	0.2902	1.4895	1.7797
12	201.96	32.36	169.9	976.6	1146.5	904.8	0.2967	1.4760	1.7727
13	205.87	30.03	173.8	974.2	1148.0	902.0	0.3025	1.4639	1.7664
14	209.55	28.02	177.5	971.9	1149.4	899.3	0.3081	1.4523	1.7604
14.696	212	26.79	180.0	970.4	1150.4	897.6	0.3118	1.4447	1.7565
15	213.0	26.27	181.0	969.7	1150.7	896.8	0.3133	1.4416	1.7549
16	216.3	24.79	184.4	967.6	1152.0	894.4	0.3183	1.4311	1.7494
17	219.4	23.38	187.5	965.6	1153.1	892.1	0.3229	1.4215	1.7444
18	222.4	22.16	190.5	963.7	1154.2	889.9	0.3273	1.4127	1.7400
19	225.2	21.07	193.4	961.8	1155.2	887.8	0.3315	1.4045	1.7360
20	228.0	20.08	196.1	960.0	1156.2	885.8	0.3355	1.3965	1.7320
21	230.6	19.18	198.8	958.3	1157.1	883.9	0.3393	1.3887	1.7280
22	233.1	18.37	201.3	956.7	1158.0	882.0	0.3430	1.3811	1.7241
23	235.5	17.62	203.8	955.1	1158.8	880.2	0.3465	1.3739	1.7204
24	237.8	16.93	206.1	953.5	1159.6	878.5	0.3499	1.3670	1.7169
25	240.1	16.30	208.4	952.0	1160.4	876.8	0.3532	1.3604	1.7136
26	242.2	15.72	210.6	950.6	1161.2	875.1	0.3564	1.3542	1.7106
27	244.4	15.18	212.7	949.2	1161.9	873.5	0.3594	1.3483	1.7077
28	246.4	14.67	214.8	947.8	1162.6	872.0	0.3623	1.3425	1.7048
29	248.4	14.19	216.8	946.4	1163.2	870.5	0.3652	1.3367	1.7019
30	250.3	13.74	218.8	945.1	1163.9	869.0	0.3680	1.3311	1.6991
31	252.2	13.32	220.7	943.8	1164.5	867.6	0.3707	1.3257	1.6964
32	254.1	12.93	222.6	942.5	1165.1	866.2	0.3733	1.3205	1.6938
33	255.8	12.57	224.4	941.3	1165.7	864.8	0.3759	1.3155	1.6914
34	257.6	12.22	226.2	940.1	1166.3	863.4	0.3784	1.3107	1.6891
35	259.3	11.89	227.9	938.9	1166.8	862.1	0.3808	1.3060	1.6868
36	261.0	11.58	229.6	937.7	1167.3	860.8	0.3832	1.3014	1.6846
37	262.6	11.29	231.3	936.6	1167.8	859.5	0.3855	1.2969	1.6824
38	264.2	11.01	232.9	935.5	1168.4	858.3	0.3877	1.2925	1.6802
39	265.8	10.74	234.5	934.4	1168.9	857.1	0.3899	1.2882	1.6781
40	267.3	10.49	236.1	933.3	1169.4	855.9	0.3920	1.2841	1.6761
41	268.7	10.25	237.6	932.2	1169.8	854.7	0.3941	1.2800	1.6741
42	270.2	10.02	239.1	931.2	1170.3	853.6	0.3962	1.2759	1.6721
43	271.7	9.80	240.5	930.2	1170.7	852.4	0.3982	1.2720	1.6702
44	273.1	9.59	242.0	929.2	1171.2	851.3	0.4002	1.2681	1.6683
45	274.5	9.39	243.4	928.2	1171.6	850.3	0.4021	1.2644	1.6665
46	275.8	9.20	244.8	927.2	1172.0	849.2	0.4040	1.2607	1.6647
47	277.2	9.02	246.1	926.3	1172.4	848.1	0.4059	1.2571	1.6630
48	278.5	8.84	247.5	925.3	1172.8	847.1	0.4077	1.2536	1.6613
49	279.8	8.67	248.8	924.4	1173.2	846.1	0.4095	1.2502	1.6597
50	281.0	8.51	250.1	923.5	1173.6	845.0	0.4113	1.2468	1.6581

Properties of Saturated Steam—Continued

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p	t	v	h	L	H	r	n_w	n_s	n_g
51	282.3	8.35	251.4	922.6	1174.0	844.0	0.4130	1.2435	1.6565
52	283.5	8.20	252.6	921.7	1174.3	843.1	0.4147	1.2402	1.6549
53	284.7	8.05	253.9	920.8	1174.7	842.1	0.4164	1.2370	1.6534
54	285.9	7.91	255.1	919.9	1175.0	841.1	0.4180	1.2339	1.6519
55	287.1	7.78	256.3	919.0	1175.4	840.2	0.4196	1.2309	1.6505
56	288.2	7.65	257.5	918.2	1175.7	839.3	0.4212	1.2278	1.6490
57	289.4	7.52	258.7	917.4	1176.0	838.3	0.4227	1.2248	1.6475
58	290.5	7.40	259.8	916.5	1176.4	837.4	0.4242	1.2218	1.6460
59	291.6	7.28	261.0	915.7	1176.7	836.5	0.4257	1.2189	1.6446
60	292.7	7.17	262.1	914.9	1177.0	835.6	0.4272	1.2160	1.6432
61	293.8	7.06	263.2	914.1	1177.3	834.8	0.4287	1.2132	1.6419
62	294.9	6.95	264.3	913.3	1177.6	833.9	0.4302	1.2104	1.6406
63	295.9	6.85	265.4	912.5	1177.9	833.1	0.4316	1.2077	1.6393
64	297.0	6.75	266.4	911.8	1178.2	832.2	0.4330	1.2050	1.6380
65	298.0	6.65	267.5	911.0	1178.5	831.4	0.4344	1.2034	1.6368
66	299.0	6.56	268.5	910.2	1178.8	830.5	0.4358	1.2007	1.6355
67	300.0	6.47	269.6	909.5	1179.0	829.7	0.4371	1.1972	1.6343
68	301.0	6.38	270.6	908.7	1179.3	828.9	0.4385	1.1946	1.6331
69	302.0	6.29	271.6	908.0	1179.6	828.1	0.4398	1.1921	1.6319
70	302.9	6.20	272.6	907.2	1179.8	827.3	0.4411	1.1896	1.6307
71	303.9	6.12	273.6	906.5	1180.1	826.5	0.4424	1.1872	1.6296
72	304.8	6.04	274.5	905.8	1180.4	825.8	0.4437	1.1848	1.6285
73	305.8	5.96	275.5	905.1	1180.6	825.0	0.4449	1.1825	1.6274
74	306.7	5.89	276.5	904.4	1180.9	824.2	0.4462	1.1801	1.6263
75	307.6	5.81	277.4	903.7	1181.1	823.5	0.4474	1.1778	1.6252
80	312.0	5.47	282.0	900.3	1182.3	819.8	0.4535	1.1665	1.6200
85	316.3	5.16	286.3	897.1	1183.4	816.3	0.4590	1.1561	1.6151
90	320.3	4.89	290.5	893.9	1184.4	813.0	0.4644	1.1461	1.6105
95	324.1	4.65	294.5	890.9	1185.4	809.7	0.4694	1.1367	1.6061
100	327.8	4.429	298.3	888.0	1186.3	806.6	0.4743	1.1277	1.6020
105	331.4	4.230	302.0	885.2	1187.2	803.6	0.4789	1.1191	1.5980
110	334.8	4.047	305.5	882.5	1188.0	800.7	0.4834	1.1108	1.5942
115	338.1	3.880	309.0	879.8	1188.8	797.9	0.4877	1.1030	1.5907
120	341.3	3.726	312.3	877.2	1189.6	795.2	0.4919	1.0954	1.5873
125	344.4	3.583	315.5	874.7	1190.3	792.6	0.4959	1.0880	1.5839
130	347.4	3.452	318.6	872.3	1191.0	790.0	0.4998	1.0809	1.5807
140	353.1	3.219	324.6	867.6	1192.2	785.0	0.5072	1.0675	1.5747
150	358.5	3.012	330.2	863.2	1193.4	780.4	0.5142	1.0550	1.5692
160	363.6	2.834	335.6	858.8	1194.5	775.8	0.5208	1.0431	1.5639
170	368.5	2.675	340.7	854.7	1195.4	771.5	0.5269	1.0321	1.5590
180	373.1	2.533	345.6	850.8	1196.4	767.4	0.5328	1.0215	1.5543
190	377.6	2.406	350.4	846.9	1197.3	763.4	0.5384	1.0114	1.5498
200	381.9	2.290	354.9	843.2	1198.1	759.5	0.5437	1.0019	1.5456
210	386.0	2.187	359.2	839.6	1198.8	755.8	0.5488	0.9928	1.5416
220	389.9	2.091	363.4	836.2	1199.6	752.3	0.5538	0.9841	1.5379
230	393.8	2.004	367.5	832.8	1200.2	748.8	0.5586	0.9758	1.5344
240	397.4	1.924	371.4	829.5	1200.9	745.4	0.5633	0.9676	1.5309
250	401.1	1.850	375.2	826.3	1201.5	742.0	0.5676	0.9600	1.5276
2947	689.0	0.05	?	0	?	0	?	0	?

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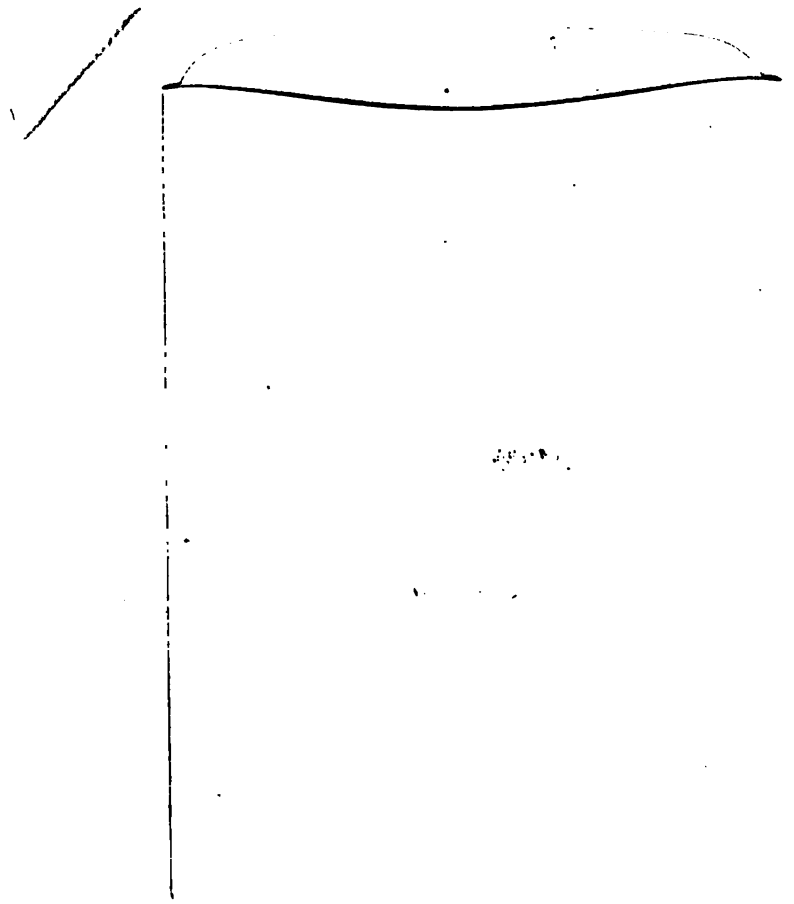

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